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CONCEPTUAL DESIGNS OF HIGH EFFICIENCY RESIDENTIAL GAS FURNACES

report prepared by

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SUMMARY

The objective of this study is to design a high efficiency residential gas-fired furnace. Conventional furnaces typically have a steady-state efficiency of approximately 80%. However, the off-cycle losses associated with cyclic operation result in seasonal efficiencies in the order of 60%, or less. The literature contains many references which detail the mechanisms governing these efficiencies. Some efforts to make modest gains in efficiency by correct furnace sizing, elimination of gas pilot lights, consideration of flue pipe dampers, etc. have been documented as well. Very little information exists in the non-proprietary literature regarding the design of high efficiency furnaces. More specifically, little information is available regarding condensing type furnaces, possibly incorporating sealed combustion systems and novel burners such as the pulse combustion design.

The patent literature does contain several designs for novel furnaces and the pertinent items are reviewed in this report.

Two new designs for high efficiency furnaces are presented in this report. One of these is an air-based furnace design incorporating a low-temperature plastic heat exchanger in the condensing section with a novel, possibly patentable, feature to permit the use of the low temperature plastic material. The other design is a hydronic-combination unit which accomplishes both the space heating and domestic hot water heating functions in the same appliance. The space heating system can be either water based or, with the provision of a blower-finned coil unit, air based.

Both furnaces rely on induced draft blowers to vent the combustion products. The ignition systems are electric. Brief consideration is given to the strategies for the control systems, ignition sequences, etc.

Preliminary cost estimates indicate that both of the designs presented here should be very competitive when compared to the cost of heating units currently available.

The two designs presented are at the development level where little additional theoretical work can be carried out. Therefore, it is recommended that both of these designs be refined in an experimental manner. It is also recommended that prototypes of both of these designs be produced and tested.

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CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1.1) BACKGROUND

The current project has as its objective a study of the design of the thermal processes of residential gas furnaces with the goal of making design changes which will result in heating appliances capable of substantially improved steady-state and seasonal efficiencies.

Until very recently, the conventional design of residential gas furnaces produced a unit with a steady-state efficiency of approximately 80%. The major portion of the energy which is not recovered is associated with the venting of high temperature flue gases[1,2]. However, in a residential installation, the furnace is called upon to function in cyclic rather than steady-state operation and as a consequence there are many other operating losses to be charged to the furnace. Thus, rather than the 80% efficiency associated with steady-state operation, the seasonal performance efficiency is probably in the range of 60% for a properly sized furnace and even lower than that if the furnace is oversized for the residential heating load[1,2]. Figure 1.1 (Reference 2) gives an approximate accounting of the various energy losses in a conventional heating system. One possible approach to the design of an efficient furnace is to study the separate loss mechanisms, make suitable adjustments or design modifications and evaluate the predicted and/or measured benefits. A considerable amount of literature exists which documents the efforts of a number of researchers and organizations over the past years in the area of furnace efficiency improvements. In particular, the proceedings from three conferences[3,4,5,6]

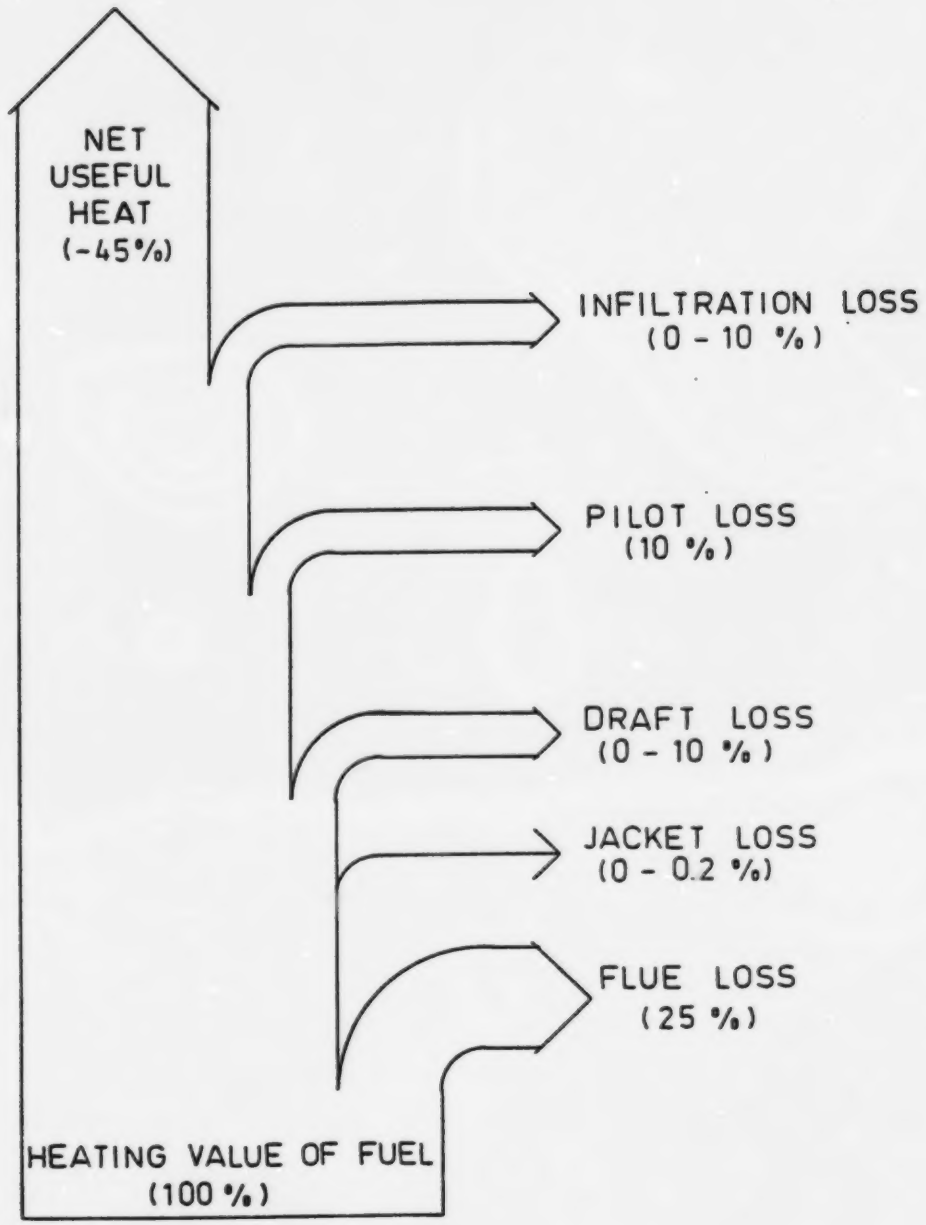

Figure 1.1 : Heat Loss Distribution from Conventional Heating System

devoted to the improvement of efficiency in HVAC equipment, components and systems summarize much of the recent work brought on by the energy awareness which the 1973 oil embargo fostered. A lot of the earlier research on burners, combustion processes, heat exchangers, etc. is found in publications[7,8,9] of the American Gas Association (A.G.A.). The Institute of Gas Technology (I.G.T.) has been active in the modelling and assessment of furnace performance[10].

Recently, a compilation and review of much of the literature specific to residential gas fired forced air heating systems was prepared by Fasken[11]. Fasken grouped the possible energy saving furnace modifications into three categories which included (a) adjustment of controls (without any additions to a conventional furnace) (b) retro-fit modifications (involving adding additional components) and (c) design alternatives. Table 1.1 outlines the specific modifications considered in each of these categories, as well as an indication of the possible energy saving based on data from the literature summarized in reference 11. Many of these modifications would be used in combinations and the results of studies of several such arrangenents are presented in Table 1.2.

1.2) OVERVIEW OF POSSIBLE DESIGN APPROACHES

All of the modifications discussed above reflect rather minor departures from conventional furnace design and in most cases are attempts to recover some of the efficiency losses associated with the cyclic performance but do not address the idea of making significant gains in the steady state efficiency. If one is willing to address the major restriction which is to maintain the combustion products at a sufficiently high temperature to prevent condensation

TABLE 1.1

HEATING SYSTEM MODIFICATIONS

Category	Range of Seasonal Efficiency Improvement (%)	References
1.0) Control Adjustment		
1.1) Air Blower On-Off Settings	6.5	2,12,13,14
1.2) Thermostat Differential	≅3	13,15,16
2.0) Retro-Fit Modifications		
2.1) Vent Damper	7 - 11	17
2.2) Electronic Ignition	5	18
2.3) Oversize Reduction	1 to 2% for every 20% reduction in oversize	19
3.0) Design Alternatives		
3.1) Sealed Combustion	4 - 11	2,17,18
3.2) Draft Hood Dilution Air	not quantified	2,13
3.3) Excess Air Reduction	10	13,15,16
3.4) Reduction of Flue Exit Temperature	-	-
3.5) Power Burners	5	18
3.6) Forced or Induced Draft Burners	14.7 - 18	14,19
3.7) Modulation Burner	10.9	14

TABLE 1.2*

COMBINATIONS OF HEATING SYSTEM IMPROVEMENTS

IMPROVEMENT COMBINATION	REFERENCE				
	19	20	21	18	12
1) Power Burner - Ignition		12%	8.8%	5%	
2) Vent Damper - Ignition		13%	9.5%	11%	10.9%
3) Sealed Combustion- Ignition				9%	5.1%
4) Forced or Induced Draft - Ignition	17%				14.5%
5) Vent Damper - Ignition - Reduced Blower Setting					19.4%
6) Forced or Induced Draft - Ignition - Reduced Blower Setting					21.4%
7) Power Burner - Ignition - Sealed Combustion				14%	

* Adapted from Fasken (reference 11)

occurring in the furnace and vent system, and one is willing to consider designs with flue gas exit temperatures below the dew point of the combustion products, then significant energy gains can be made.

Still another major design change which originated many years ago but has surfaced again recently is the use of a pulse combustion system in place of the more conventional atmospheric or power burners.

Recently condensing furnace designs both with[22,23] and without[24] pulse combustion have been introduced to the commercial market. However, the non-proprietary literature contains very little design information for condensing furnaces. A recent seminar[25] under the sponsorship of the U.S. Department of Energy, Brookhaven National Laboratory and the Gas Research Institute outlined the extent of an R and D project on condensing heat exchangers currently under way at three laboratories (Battelle-Columbus, The Canadian Gas Research Institute (CGRI) and the American Gas Association (AGA) Laboratories) but was directed predominantly at the materials evaluation phase, not at furnace design.

If one considers the design of a condensing furnace, since the flue products have been cooled sufficiently that natural draft venting using a chimney is no longer feasible, the design probably includes forced or induced draft venting (or pulse combustion), and electronic ignition. The design may or may not incorporate a sealed combustion system and the overall performance becomes very much less sensitive to such parameters as percent excess air, degree of furnace oversizing, blower on-off settings, etc. Vent dampers and draft hoods are no longer required. As a consequence, much of the above literature on furnace efficiencies, although important for existing installations

and retro-fit modifications is of only minor relevance to the design of a high efficiency condensing furnace.

1.3) PROJECT OVERVIEW

This report presents design details of two high efficiency furnace designs. One unit is an air based condensing furnace and the other is a hydronic condensing system which supplies domestic hot water plus space heating.

Each of these units incorporates an induced draft flue gas exhaust as well as electronic ignition. Therefore, it is expected that the seasonal efficiency will be very close to the steady state efficiency in each case.

A review of the patent literature concerning high efficiency gas fired furnaces was carried out. It is the opinion of the authors that neither of the designs presented in this study interfere with any patents found during this search. A discussion of the findings of the patent literature search is presented in Chapter 2.

CHAPTER 2

PATENT LITERATURE REVIEW

2.1) PATENT REVIEW PROCEDURE

During the course of this study, a review of the patent literature was undertaken. The overall objective of this search was to uncover any patents that describe gas fired domestic heating processes and/or equipment. Having completed this search it was possible to judge more readily whether or not the furnace designs being developed constituted an infringement upon any previous patents.

The patent review was executed by means of two computer searches plus a manual search. The computer aided patent searches included examination of the U.S. Gazette patent listing and the American Petroleum Institute (API) patent listing, the former being much more rewarding. In addition, a manual search of the U.S. Gazette patent literature was carried out.

These patent searches cover the fourteen year time period prior to the date of the search (August, 1981). Excluded from this period is the most recent year in the case of the manual search and the most recent month in the case of the computer search. Patent information concerning the more recent time periods mentioned above was still being processed and was unavailable.

The initial stage of the patent review involved a search of the patent index listings using various key words relevant to the domestic heating process. Ninety-four patent numbers were revealed. Subsequent stages of the

search included the review of material corresponding to each of the patent numbers. The patent literature was reviewed first by title, and then in progressively more detailed steps including one line descriptions, short abstracts, full abstracts and full patents. Certain patents were removed from the search at each of the various steps in the review process as it became evident that these patents were of little or no use. Complete patents corresponding to the most interesting abstracts were examined carefully. Copies of the most interesting patents are included in Appendix B. An overall summary of the patent literature reviewed is included in Appendix A.

2.2) PATENTS OF INTEREST

Two patents included in Appendix B are of particular interest. Certain aspects of the inventions presented in these patents resemble some aspects of the heating system designs presented in this report.

First, the hot air heater described in U.S. Patent Number 4,069,807 is a process by which flue gases are saturated and sufficiently cooled through direct contact with water such that they can then be further cooled in a heat exchanger made of low service temperature plastic. It is further specified that this plastic heat exchanger has between 3 and 10 square feet of air contact area per 1,000 BTU/hr of burner input. The air based system described in this report incorporates a condensing heat exchanger built from low service temperature plastic. However, the process involved in this air based furnace does not involve direct contact between flue gas and water and uses less than 2 square feet of heat exchanger air side contact area per 1,000 BTU/hr of burner input. The water based system described in this report neither

involves the use of low temperature plastic nor incorporates direct contact between flue gas and water.

Second, the heat exchanger described in U.S. Patent Number 4,171,089 is constructed such that it communicates with combustion air drawn from outdoors, house return air and combustion products. The flue gas is cooled sufficiently in this heat exchanger such that condensation takes place. The heat exchanger passages which transport flue gas are oriented such that the flue gas and the condensate flow in opposite directions. As a result of this arrangement, the claim is made that this heat exchanger is self cleaning. The air based furnace presented here also transfers heat from the flue products to return air and combustion air but it incorporates neither a reversed condensate flow nor the same flue gas flow paths shown in the patent mentioned above.

The remainder of the patents included in Appendix B describe a pulse combustion system for heating air as well as various other heat exchanger designs.

It is the opinion of the authors that neither of the designs presented in this study interfere with any patents found during this patent literature review.

CHAPTER 3

GENERAL FURNACE CALCULATIONS

3.1) THE COMBUSTION PROCESS

Before any specific furnace design is examined, it is instructive to present several general calculations that quantify the various processes that take place on the combustion - flue gas side of a gas fired furnace.

Throughout these calculations natural gas will be treated as if it consisted of 100% methane, CH_4. This simplification is a common practice. A general chemical expression representing the combustion of methane is shown by equation 3.1.

$$CH_4+2\phi(O_2+3.76N_2) \rightarrow CO_2+2fH_2O_\ell+2(1-f)H_2O_v+7.52\phi N_2+2(\phi-1)O_2 \qquad (3.1)$$

where: ϕ quantifies the amount of excess air present (% theoretical air = 100ϕ and % excess air = $100(\phi-1)$)

f is defined as the portion of the water vapour in the products that has condensed. In a conventional furnace, f=0.

Considering the furnace as a simple control volume, as shown in Figure 3.1, it is apparent that the condensed fraction of water vapour in the flue products, f, can be determined as a function of flue gas exhaust temperature (or rate of heat removal, $\dot{Q}$) and the amount of excess air supplied. Examining equation 3.1 it can be seen that the partial pressure of the water vapour in the flue products, P_{H_2O}, is as shown in equation 3.2.

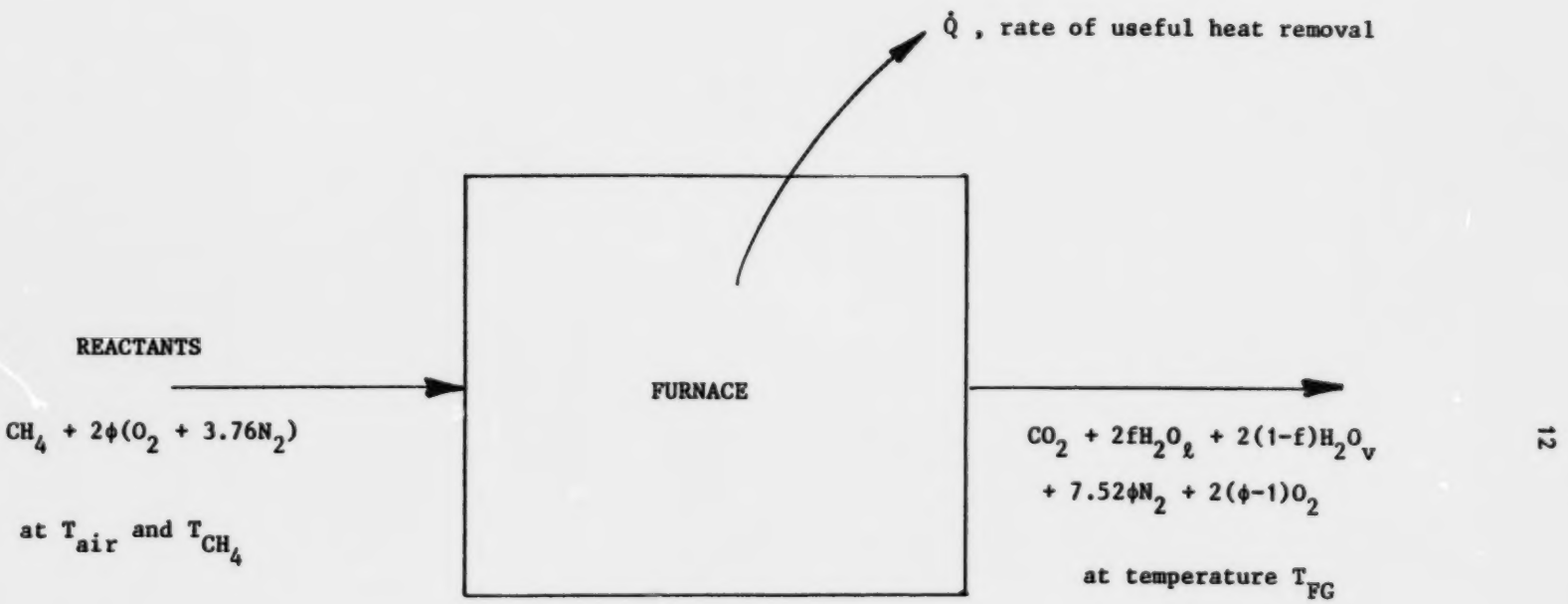

Figure 3.1: The Furnace Analysed as a Control Volume

$$P_{H_2O} = P_{TOT} \cdot \frac{2(1-f)}{1 + 2(1-f) + 7.52\phi + 2(\phi-1)} \qquad (3.2)$$

Throughout this investigation it was assumed that all processes take place within the furnace at one atmosphere pressure. Thus, P_{TOT} = 1 atmosphere.

For cases when the flue gas temperature is below the dew point, rearranging equation 3.2 and substituting the saturation pressure of water, P_{SAT}, in place of the partial pressure of water, P_{H_2O}, enables one to solve for the fraction of water condensed as shown in equation 3.3.

$$f = 1 - \frac{9.52(\phi-1)}{2(\frac{P_{TOT}}{P_{SAT}} - 1)} \qquad (3.3)$$

Having solved equation 3.3 the fraction water condensed from the flue gas, f, is shown in Figure 3.2 as a function of both excess air and the temperature corresponding to the various values of water vapour saturation pressure, P_{SAT}. P_{SAT} can be found as a function of temperature in standard thermodynamic saturation steam tables.

The dew point temperature of the flue gas, T_{DP}, can be found by examining Figure 3.2 at f=0 or by setting f=0 in equation 3.2. The result of this substitution is:

$$P_{SAT} = \frac{2}{9.52 + 1} \cdot P_{TOT} \qquad (3.4)$$

Having solved equation 3.4 for P_{SAT}, one can then use saturation steam

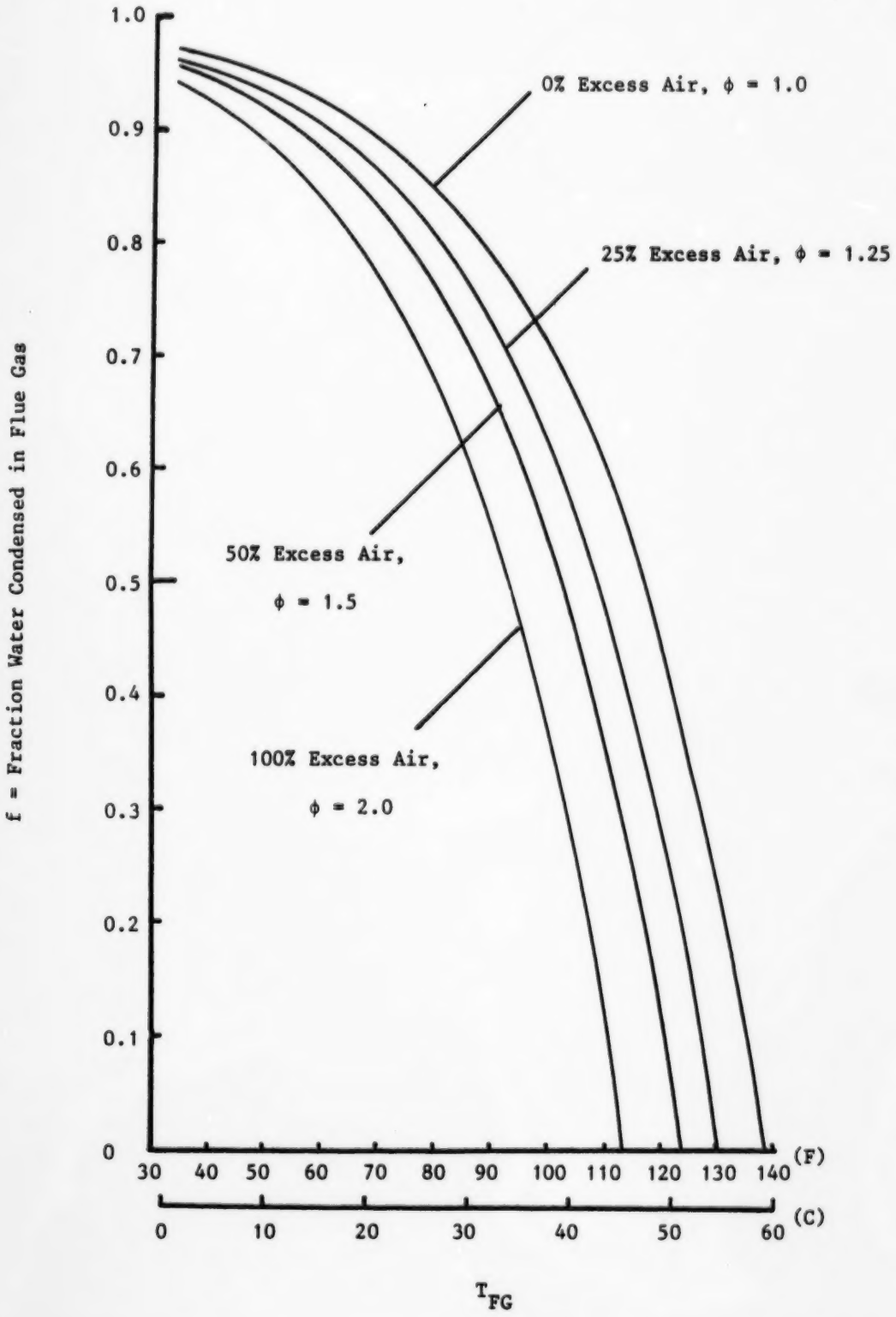

Figure 3.2: Fraction Water Condensed vs. Flue Gas Temperature

table values to obtain the dew point temperature corresponding to P_{SAT}. This is shown as a function of percent excess air in Figure 3.3.

Performing an energy balance on the furnace control volume of Figure 3.1 yields equation 3.5.

$$\hat{h}_{CH_4} + 2\phi\hat{h}_{O_2} + 7.52\phi\hat{h}_{N_2} = Q + \hat{h}_{CO_2} + 2f\hat{h}_{H_2O_\ell} + 2(1-f)\hat{h}_{H_2O_v} + 7.52\phi\hat{h}_{N_2} + 2(\phi-1)\hat{h}_{O_2} \qquad (3.5)$$

where: $\hat{h}_i$ = the enthalpy per mole of the i^{th} component (e.g. CH_4, O_2, etc.)

In order to make use of equation 3.5, it was assumed that the methane enters the furnace at +21.1 C (70 F) and combustion air is drawn from outdoors at -1.1 C (30 F). Under these conditions the amount of thermal energy derived from the cooling of the flue products, per mole of methane consumed, was calculated by solving equation 3.5. Figure 3.4 shows the resulting data as a function of flue gas temperature and percent excess air. Figure 3.5 shows the same data in greater detail in the low temperature condensing range. Equation 3.5 was solved using tabulated enthalpy data[26]. The values of fraction water condensed, f, were calculated as a function of temperature and percent excess air in the same manner outlined earlier for the construction of Figure 3.2. The efficiency scale shown in Figures 3.4 and 3.5 is based on equation 3.6.

$$\eta = \frac{\dot{Q}/\dot{n}_{CH_4}}{HHV_{CH_4}} \qquad (3.6)$$

where: $\dot{n}_{CH_4}$ is the molal flow of methane per unit time

$\dot{Q}$ is the thermal energy release per unit time

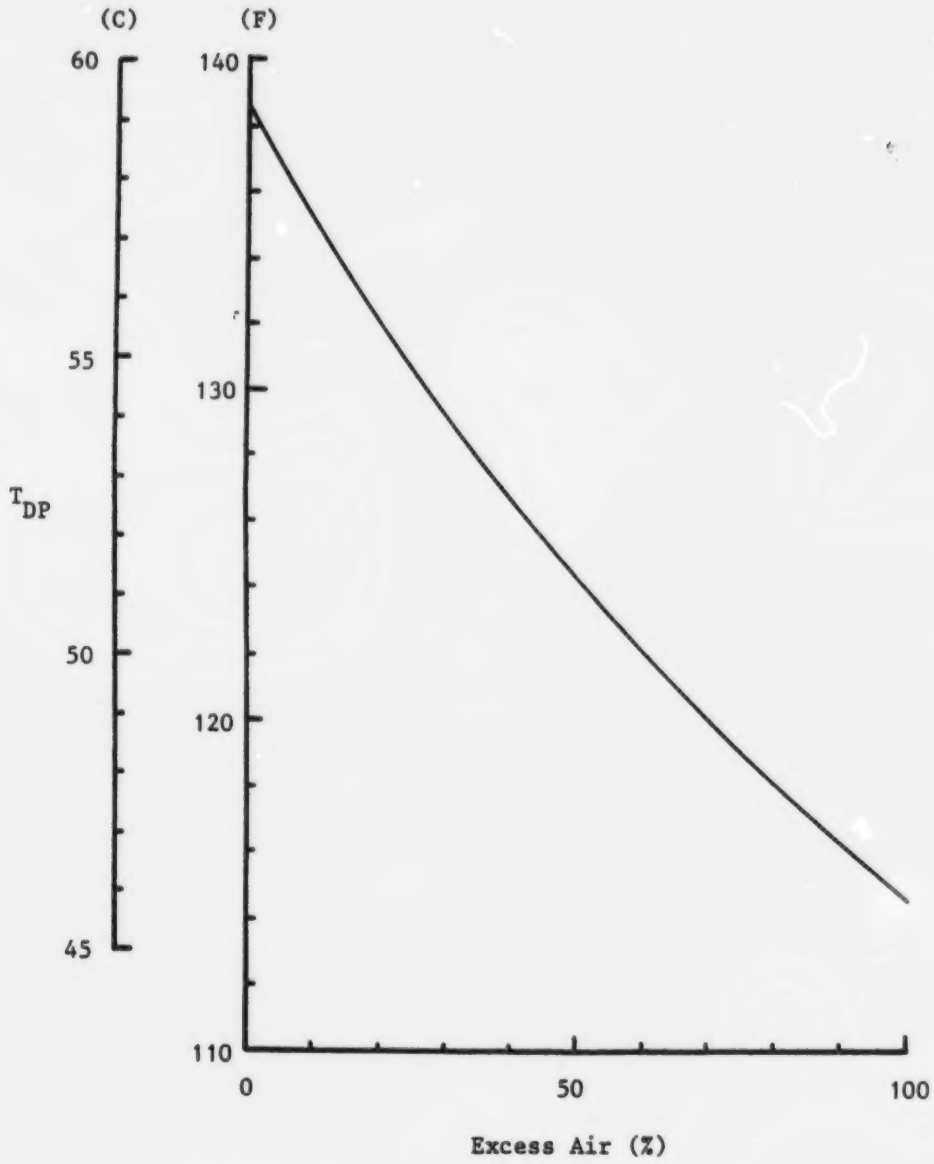

Figure 3.3: Flue Gas Dew Point Temperature vs. % Excess Air

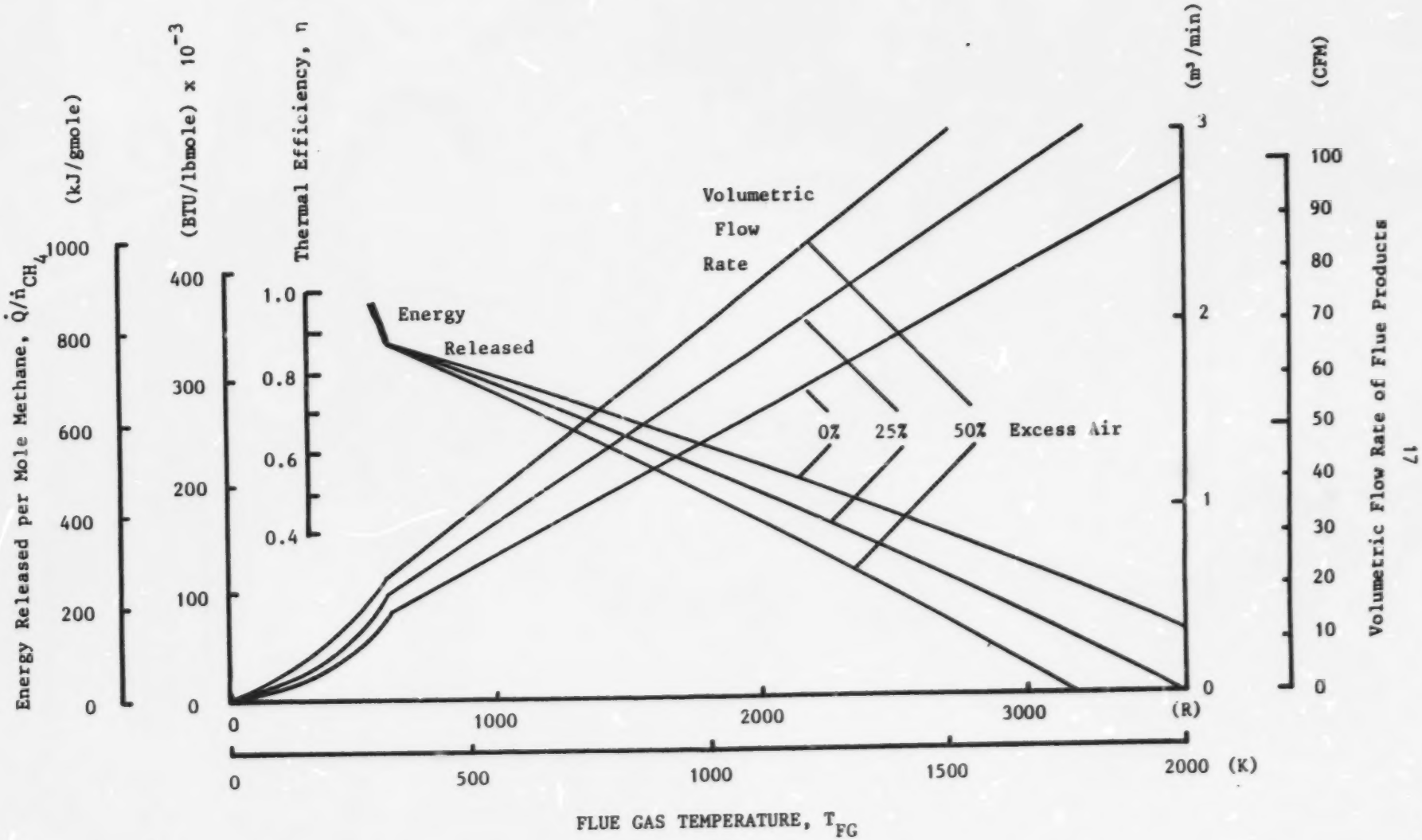

Figure 3.4: Energy Released and Volumetric Flow Rate vs. Flue Gas Temperature

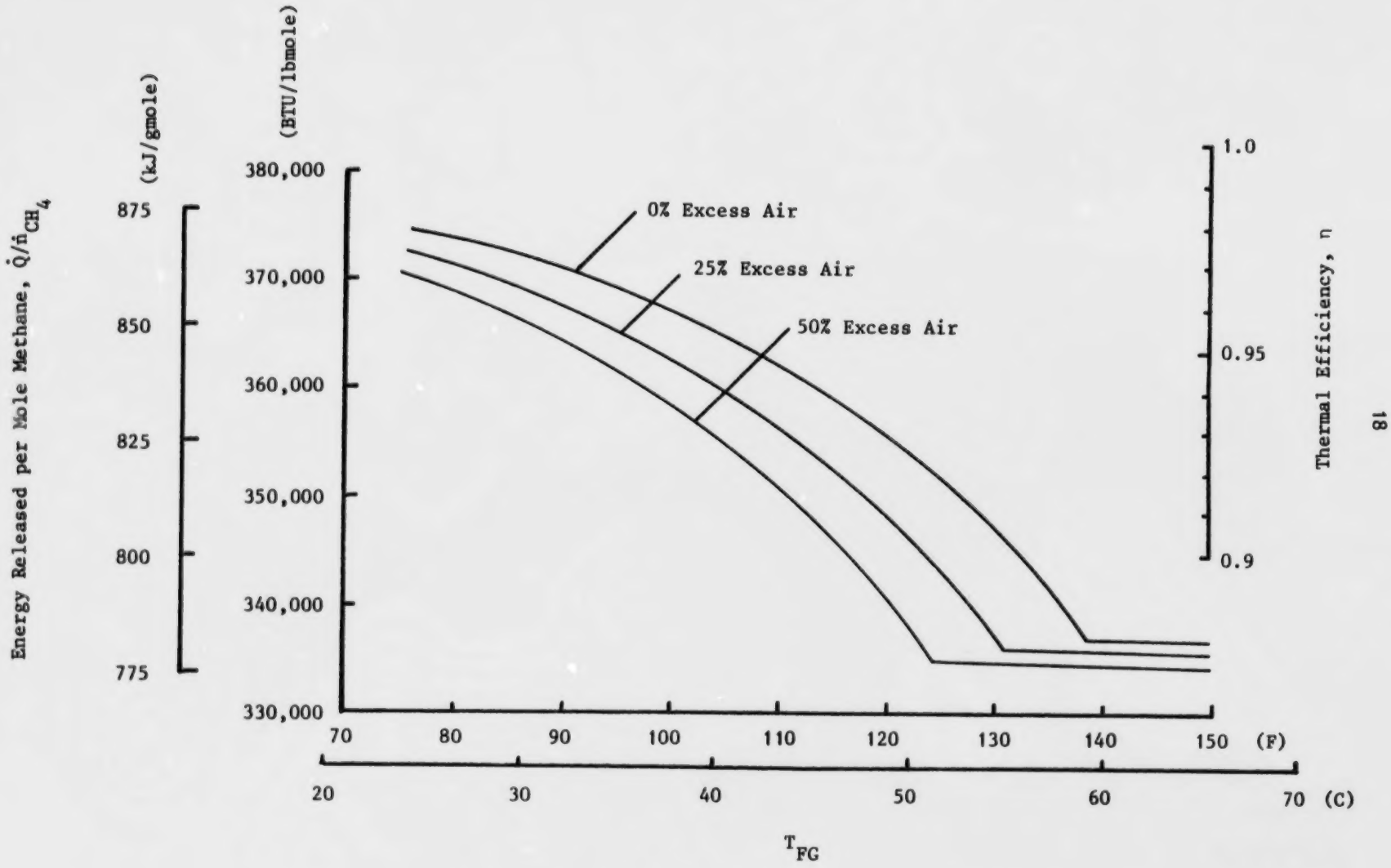

Figure 3.5: Thermal Energy Release vs. Flue Gas Temperature

HHV_{CH_4} = the higher heating value of methane ≅ 383 x 10^3 BTU/lbmole
= 890 kJ/gmole

Thus, by measuring the bulk temperature and CO_2 content (to determine excess air) of the flue gas leaving a furnace it is possible to estimate the thermal efficiency of that furnace by using Figure 3.4 or Figure 3.5.

3.2) DESIGN CONDITIONS AND FLOW RATES

In order to size the various heat exchangers studied during this project it was decided that the furnace input should be fixed at $\dot{Q}_{in}$=23.44kW (80,000 BTU/hr). It is felt that this is a popular size and that the heating load of the great majority of residential buildings could be met by a high efficiency furnace of this input. There would be little problem in considering furnaces either larger or smaller than the size arbitrarily chosen for these calculations.

During burner operation, the molal flow rate of methane is:

$$\dot{n}_{CH_4} = \frac{\dot{Q}_{in}}{HHV_{CH_4}} = 0.095 \text{ kgmole/hr} = 0.209 \text{ lbmole/hr} \qquad (3.7)$$

The mass flow rate of methane is then:

$$\dot{m}_{CH_4} = \dot{n}_{CH_4} \cdot \hat{M}_{CH_4} = 1.52 \text{ kg/hr} = 3.35 \text{ lbm/hr} \qquad (3.8)$$

where the molecular weight of methane is: $\hat{M}_{CH_4}$ = 16.04 kg/kgmole
= 16.04 lbm/lbmole

Similarly, the flow rate of combustion air is given by:

$$\dot{n}_{air} = \left(\frac{\dot{n}_{air}}{\dot{n}_{CH_4}}\right)\dot{n}_{CH_4} \tag{3.9}$$

where: $$\frac{\dot{n}_{air}}{\dot{n}_{CH_4}} = 9.52\phi \tag{3.10}$$

For the case of 35% excess air or ϕ=1.35, the molal flow rate of combustion air is:

$$\dot{n}_{air} = 1.22 \text{ kgmole/hr} = 2.69 \text{ lbmole/hr}$$

The mass flow rate of air under this condition is:

$$\dot{m}_{air} = \dot{n}_{air} \cdot \hat{M}_{air} = 35.3 \text{ kg/hr} = 77.9 \text{ lbm/hr} \tag{3.11}$$

Therefore, the total mass flow rate of reactants (or combustion products) is:

$$\dot{m} = \dot{m}_{CH_4} + \dot{m}_{air} = 36.0 \text{ kg/hr} = 79.4 \text{ lbm/hr} \qquad @\ \phi=1.35 \tag{3.12}$$

Using the perfect gas approximation, the volumetric flow rate of the combustion air, $\dot{V}_{air}$, is:

$$\dot{V}_{air} = \frac{\dot{m}_{air}\, T_{air}}{p} R_{air} \tag{3.13}$$

where R_{air} is the gas constant for air.

Substituting;

$$T_{air} = 272.1 \text{ K} = 490 \text{ R}$$

$$P = 1 \text{ atm}$$

yields,

$$\dot{V}_{air} = 27.3 \text{ m}^3/\text{hr} = 16.0 \text{ CFM} \qquad @ \ \phi=1.35$$

Calculation of the volumetric flow rate of the flue gas, $\dot{V}_{FG}$, is somewhat more complicated. Again,

$$\dot{V}_{FG} = \frac{\dot{m}_{FG} T_{FG}}{P} R_{FG} \tag{3.14}$$

where;

a) $$\dot{m}_{FG} = \dot{m}_{air} + \dot{m}_{CH_4} - f\dot{m}_{H_2O} \tag{3.15}$$

The term $f\dot{m}_{H_2O}$ is the mass flow rate of condensed water vapour in the flue gas. Knowing that:

$$\dot{m}_{H_2O} = \hat{M}_{H_2O}\dot{n}_{H_2O} = \hat{M}_{H_2O}\left(\frac{\dot{n}_{H_2O}}{\dot{n}_{CH_4}}\right)\dot{n}_{CH_4} = \frac{\hat{M}_{H_2O}}{\hat{M}_{CH_4}}\left(\frac{\dot{n}_{H_2O}}{\dot{n}_{CH_4}}\right)\dot{m}_{CH_4} = 2.25\dot{m}_{CH_4}$$

equation 3.15 becomes;

$$\dot{m}_{FG} = \dot{m}_{air} + \dot{m}_{CH_4}(1 - 2.25f) \tag{3.16}$$

b) T_{FG} = flue gas temperature

c) P = 1 atmosphere

d) R_{FG} = combustion products gas constant

$$R_{FG} = \frac{\mathcal{R}}{\hat{M}_{FG}} \tag{3.17}$$

where $\hat{M}_{FG}$ is the molecular weight of the flue gas and $\mathcal{R}$ is the universal gas constant.

$$\hat{M}_{FG} = \frac{\Sigma \hat{M}_i \dot{n}_i}{\Sigma \dot{n}_i} \tag{3.18}$$

The subscript "i" refers to the various components of the flue gas. Referring to equation 3.1;

$$\Sigma \dot{n}_i = 1 + 2 - 2f + 7.52\phi + 2(2\phi - 2) = 1 + 9.52\phi - 2f$$

and,

$$\Sigma \hat{M}_i \dot{n}_i = 44.01 + 18.016(2-2f) + 28.02(7.52\phi) + 32(2\phi-2)$$
$$= 16.042 + 274.7\phi - 36.032f$$

so;

$$R_{FG} = \mathcal{R}\left(\frac{1 + 9.52\phi - 2f}{16.042 + 274.7\phi - 36.032f}\right) \tag{3.19}$$

Using the expressions derived, the resulting data are presented in Table 3.1. Curves representing these flow rate data are also shown in Figure 3.4. It is interesting to note that the gas constant, R_{FG}, differs only slightly from that of air.

TABLE 3.1

VOLUMETRIC FLOW RATE OF FLUE GAS*

vs. FLUE GAS TEMPERATURE AND EXCESS AIR

T_{FG} (R)	ϕ = 1.0	ϕ = 1.25	ϕ = 1.5	f	R_{FG} (ft-lbf/lbm-R)
4000	107.4	-	-	0	55.90
4000	-	131.7	-	0	55.45
4000	-	-	156.0	0	55.14
2000	53.7	-	-	0	55.90
2000	-	65.8	-	0	55.45
2000	-	-	78.0	0	55.14
598.6	16.1	-	-	0	55.90
590.8	-	19.5	-	0	55.45
584.4	-	-	22.8	0	55.14
	14.3	-	-	0.45	54.14
580	-	18.3	-	0.29	54.54
	-	-	22.2	0.14	54.79
	13.0	-	-	0.71	53.02
560	-	16.7	-	0.62	53.43
	-	-	20.3	0.54	53.68
	12.2	-	-	0.85	52.40
540	-	15.6	-	0.81	52.79
	-	-	19.0	0.76	53.04

* Flow rates shown in Table 3.1 are in units of Cubic Feet per Minute. S.I. units are shown in Figure 3.4 where curves representing these data are presented.

3.3) COMBUSTION AIR PREHEAT

When considering the use of an additional condensing heat exchanger for the purpose of preheating combustion air (drawn from outdoors), it is of interest to be able to answer questions such as:

How much can the flue gases be cooled before they are no longer useful for combustion air preheat?

At what rate will the combustion air temperature rise with respect to the temperature drop of the combustion products?

The answer to these questions is not immediately obvious because consideration must be made for the condensation taking place on the flue gas side. In order to estimate the relative temperature change of the two gas flows, dT_a/dT_{FG}, the following analysis is presented.

A sample of air on the combustion air side gains thermal energy, dq, in a period of time.

$$dq = n_a \hat{C}_{p,a} dT_a \tag{3.20}$$

where, n_a is the number of moles in the air sample.

dT_a is the temperature rise of the air sample.

This same amount of energy, dq, is given up by the flue gases.

This can be denoted by:

$$dq = -n_{FG}\hat{C}_{p,FG}dT_{FG} + \hat{h}_{fg}dn_{H_2O} \tag{3.21}$$

The first term of equation 3.21 denotes sensible energy given up by n_{FG} moles of flue gas. The second term represents the latent heat released by the condensation of dn_{H_2O} moles of water ($\hat{h}_{fg}$ is the molal heat of condensation.).

Equating equations 3.20 and 3.21 and converting the molal quantities to molal flow rates, it can be shown that:

$$\frac{dT_a}{dT_{FG}} = \frac{\dot{n}_{CH_4}}{\dot{n}_a}\left(\frac{\hat{h}_{f,g}}{\hat{C}_{p.a}} \cdot \frac{\dot{n}_{H_2O}}{\dot{n}_{CH_4}} \cdot \frac{df}{dT_{FG}} - \frac{\dot{n}_{FG}}{\dot{n}_{CH_4}} \cdot \frac{\hat{C}_{p,FG}}{\hat{C}_{p,a}}\right) \tag{3.22}$$

where:

df/dT_{FG} is the slope of the plot shown in Figure 3.2.

Assuming that:

$$\frac{\hat{C}_{p,FG}}{\hat{C}_{p,a}} = 1$$

$$\frac{\hat{h}_{fg}}{\hat{C}_{p,a}} = 1495\ C = 2691\ F$$

and noting from equation 3.1 that:

$$\frac{\dot{n}_{H_2O}}{\dot{n}_{CH_4}} = 2$$

$$\frac{\dot{n}_{FG}}{\dot{n}_{CH_4}} = 1 + 9.52\phi$$

$$\frac{\dot{n}_a}{\dot{n}_{CH_4}} = 9.52\phi$$

equation 3.22 becomes:

$$\frac{dT_a}{dT_{FG}} = \frac{1}{9.52}\left(5382 \cdot \frac{df}{dT_{FG}} - 1 - 9.52\phi\right) \qquad (3.23)$$

where df/dT_{FG} is in units of F^{-1}.

Now assume that the temperature change of the flue gas in the combustion preheat section is relatively small. Then, it is reasonable to evaluate df/dT_{FG} at a representative point and to use that value over a range of combustion air temperatures and a small range of flue gas temperatures.

Assume flue gas is taken from a condensing furnace at T_{FG} = 37.8 C (100 F) and combustion has taken place with 35% excess air (ϕ=1.35). Referring to Figure 3.2, f=0.58 (58% of the water has already been condensed.). The value of df/dT_{FG} is measured graphically from Figure 3.2, $df/dT_{FG} = 1/74\ F^{-1}$. Using equation 3.23, $dT_a/dT_{FG} = -4.6$.

Thus, if this flue gas is used to preheat combustion air, the air temperature

will rise by approximately 4.6 degrees for each degree that the temperature of the flue gas drops. If combustion air must be heated through a 27.8 C (50 F) temperature rise, the flue gas temperature will drop from 37.8 C (100 F) to about 31.7 C (89 F) (Note that the assumption concerning the relatively small drop in flue gas temperature is reasonable.).

Reading Figure 3.5, it can be seen that the addition of this combustion air preheat would have increased the overall furnace efficiency by roughly one percentage point. Therefore, the savings presented to the furnace user, by the addition of this heat exchanger, will be about 1% of his heating bill.

The use of combustion air preheat also provides a practical benefit for furnace operation. By reducing the flue gas temperature as low as possible before the flue gas leaves the furnace (and removing as much water as possible) the problem of ice formation at or near the outside wall exhaust port is correspondingly reduced. In fact, if the flue gas is cooled below room temperature at the preheat exchanger and then warmed as it passes through the exhaust piping its relative humidity will have dropped below the saturation level. This could reduce the possibility of condensation in the blower housing and eliminate icing on the exhaust port parts depending upon the particular configuration in question.

Whether the initial expenditure associated with the combustion air heat exchanger is economically sound depends upon many factors including the cost of energy, now and in the future, as well as the importance of being able to reduce the amount of water vapour in the flue gas exhaust. Therefore, a truly useful economic prediction is beyond the scope of this report. However,

it is apparent that combustion air preheat will be viable only if it appears to be possible with a small marginal materials cost and minimal additional manufacturing complexity.

CHAPTER 4

GENERAL FURNACE DESIGN CONSIDERATIONS

4.1) CONTROL STRATEGY

A major component of the satisfactory design of any space heating system is the design of a control system that will insure safe and reliable operation. Such systems are built into many household appliances. As a result, many parts such as temperature switches, pressure switches, time switches, etc. are readily available. In fact, the control systems that are expected to be suitable for use with the heating systems presented here have already been incorporated, tested and approved for use on heating equipment currently being marketed.

Both of the heating units described in this report incorporate an induced draft system for flue gas movement plus electronic ignition. Therefore, much of their control systems will be of a common design. For instance, upon a call for burner operation the sequence of control events might be as follows:

a) Start flue gas blower

b) Prove flow on flue gas side. This can be done by using a pressure tap at a suitable location in the flue gas path.

c) Allow blower to purge combustion side for ½ minute

d) Initiate pilot flame ignition

e) Prove pilot flame

f) Supply gas to burner

A necessary safety feature of any direct vent appliance is that it should shut itself off in the event of exhaust pipe blockage. It is possible to detect blockage using the same pressure tap that is used for proving flue gas flow by detecting a higher than normal pressure level. The pressure level of the no-flow situation depends upon the choice of exhaust blower.

A blockage situation could also be detected by a temperature switch located in the mouth of the combustion chamber area. In this manner the spillage of hot flue gas could be sensed and the unit shut down. However, spillage would not be detected during steps "a" through "e" of a start up sequence if a spillage switch alone was used to detect blockage. In this case, gas would be supplied to the burner without a proper purge of the combustion changer.

As is the case with a conventional gas fired furnace, over-temperature protection must be provided for the heat exchanger(s). In the case of the air based unit presented here two temperature sensors will be required. Over-temperature protection can be provided for the hydronic unit by the use of standard relief valves.

4.2) TREATMENT OF CONDENSATE

One problem common to all condensing type heating units is that the condensate produced is mildly acidic (nominal ph of 3.5.to 4.0)[27]. In

general, municipal codes specify that nothing be discharged to the sewer system with a ph higher than 10 or lower than 5 [27]. In order to meet this requirement the furnace condensate must first be treated to raise its ph level.

Probably the most simple method of condensate treatment is to pass the condensate through a bed of basic material. Possible bed materials include soda lime and dolomite chips[27]. One condensing furnace unit currently on the market uses a bed of magnesium oxide. The cost of this treatment unit is approximately $40. However, it has been indicated that much of this cost is associated with the purchase of the container and that this cost could be substantially reduced*.

Care must be taken that the residence time of the condensate is such that the final ph level is between 5 and 10.

4.3) FLUE GAS AND RETURN AIR BLOWERS

The pressure drop through the combustion side of a conventional furnace is typically quite small. In fact, it is small enough that the flue gas can be exhausted by means of a natural draft mechanism. However, the condensing furnace poses a different problem. A condensing furnace removes a larger portion of the thermal energy in the flue gas through the use of additional heat exchange area. This additional heat transfer takes place predominantly by

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* Personal communication with E.C. (Ted) Clare concerning the Clare Brothers Megasave 1 furnace design.

convective heat transfer and results in a corresponding increase in flue gas pressure drop (refer to the Reynolds Analogy, reference 28).

As shown in Figure 3.4, the volumetric flow rate of flue gas is quite small at the exit of a condensing furnace. This flow rate will be less than 0.5 m^3/hr (18 CFM) for the furnace size considered in this study (23.4 kW (80,000 BTU/hr) input). Therefore, flue gas pressure drops of up to several inches of water guage do not present blower power problems. Clearly, the most prevalent problem will be the task of finding a blower with the proper performance characteristics once the pressure drop of a particular furnace has been experimentally quantified.

The additional heat exchange area mentioned above will also increase the pressure drop on the return air side of the furnace. Again, blower power is not expected to present a problem. It is expected that the air side blower requirements for the heating system designs presented in this report can be handled with a blower similar to, or the same as, furnace blowers currently in use as these blowers are normally operated well below their maximum capacity*.

Throughout the heat exchanger design procedures presented in Chapters 5 and 6, pressure drop calculations and/or results are shown whenever reasonably possible.

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* Personal communication with E.C. (Ted) Clare.

CHAPTER 5

THE HYDRONIC COMBINATION UNIT

5.1) THE CONCEPTUAL DESIGN

The design of a combination hydronic heating system was pursued as part of this study in recognition of the requirements of the portion of the space heating market that will use water as an intermediate heat transfer fluid. In addition, it is felt that there may also be a demand for a single, direct vent, high efficiency, space heating and domestic hot water unit that can be used for forced air type space heating (depending upon the expenditure involved for such a system). Therefore, the system analysed in this report includes the finned coil heat exchanger unit that would be used for the forced air space heating application.

Initially, possible methods of cooling the flue gas were considered. It was recognized that the flue gas and water flows must be arranged in an overall counterflow arrangement in order to consistently be able to cool the flue gas below its dew point. This requirement suggested a sectioned heat exchanger or a multiple heat exchanger arrangement. It was also recognized that this concept corresponds well to the shift from radiation to convection dominated heat transfer and to the shift from non-corrosive to corrosive flue products on the combustion side as the flue gas is progressively cooled. In fact, this situation is well suited to a two heat exchanger system, each one being designed for a specific task.

At this point it was apparent that the requirements of the first of these

two heat exchangers could be met using a standard domestic hot water heater. Therefore, it was decided that a combination heating unit capable of supplying hot water plus residential space heating would be designed and that this design would incorporate the standard storage type hot water heater as its burner section and primary heat exchanger. A schematic diagram outlining the operation of this unit is shown in Figure 5.1.

The major feature of this combination unit is that it is based on the very reasonably priced and reliable gas fired water heater design that has been on the market for some time. Flue gases drawn from the conventional water heater are passed through a set of tubes that are contained within a water preheat tank. In this second tank the flue gases are cooled into the condensing regime and are then exhausted using a small induced draft blower. Water enters this preheat tank from either the finned coil used for space heating or from the supply mains, depending on whether the unit is being used as a furnace or as a water heater. The partially heated water is routed from the preheat tank to the main water heater tank. Water leaving the main tank is used for domestic hot water or flows to the finned coil depending on whether the unit is operating as a water heater or as a furnace.

When the unit operates in the furnace mode a small pump circulates water through a loop comprised of the two water tanks plus the finned coil heat exchanger. When the unit operates as a hot water heater a check valve prevents mains water from bypassing the heating tanks via the finned coil unit.

As in the operation of a standard domestic hot water heater, burner operation is controlled by a thermostat which monitors the temperature near the

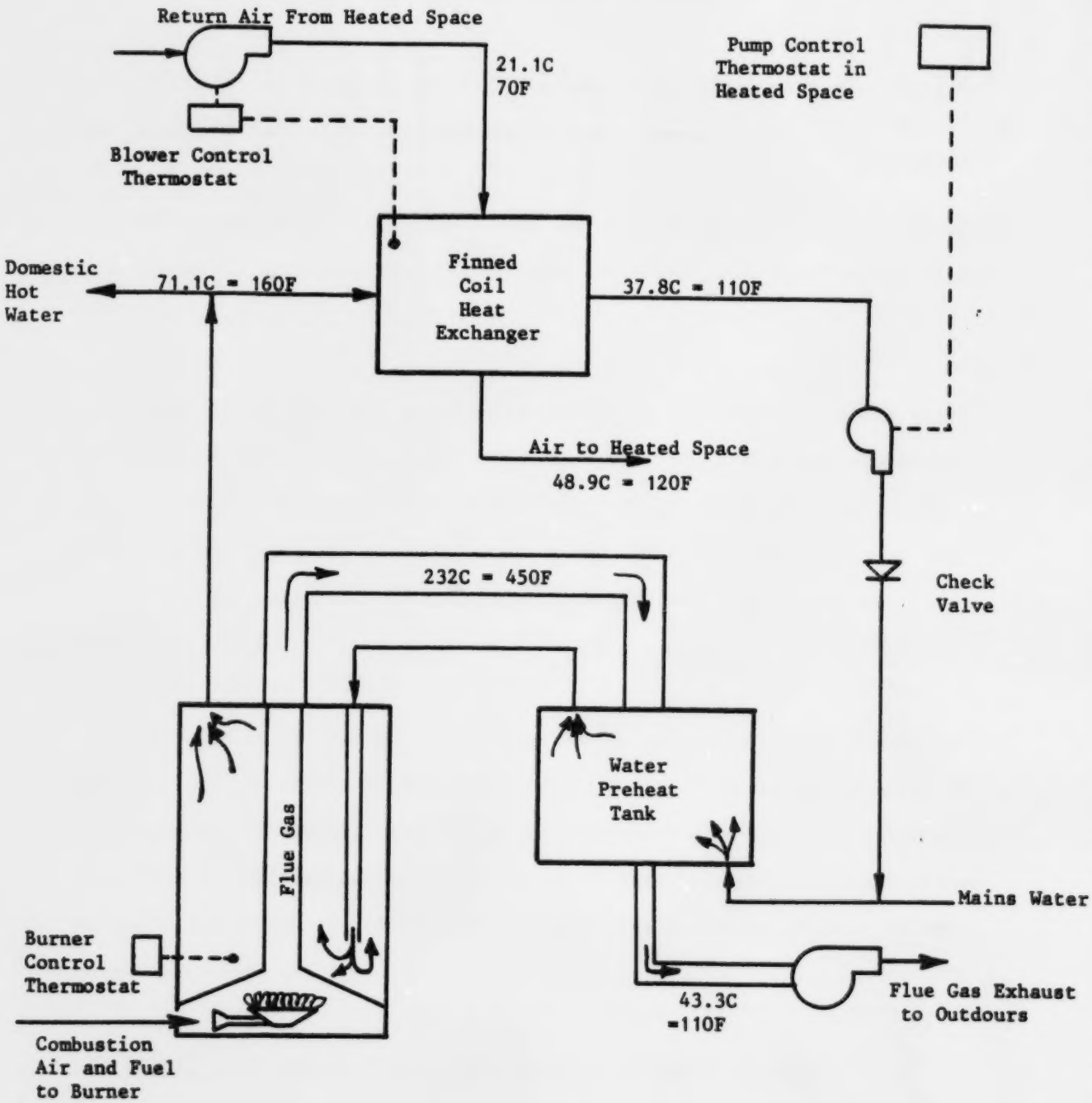

Figure 5.1: The Hydronic Combination Unit

bottom of the main heater tank. In this instance the only difference that might be perceived is that an electronic ignition is necessary.

Operation of the space heating loop water pump is controlled directly by the thermostat located in the residential heated space.

The blower which circulates air to and from the heated space is controlled by a thermostat which monitors the temperature of the water entering the finned coil heat exchanger.

Thus, a call for heat will initially turn on the water pump which will circulate hot water to the finned coil. This will trip the blower thermostat and the required heated air will be provided. Cooler water entering the hot water tank will trip the burner thermostat and additional energy will be delivered to the hot water tanks until the water heater set point temperature has again been reached.

It is expected that hot water draws during the space heating operation of this unit will result in no detrimental effects since the storage capacity of a 24.4 kW (80,000 BTU/hr) input gas fired water heater is typically twice as large as the usual residential size unit. For instance, the A.O. Smith Conservationist, model PGC-75, (80,000 BTU/hr input) has a storage capacity of 284 liters (62.5 imp. gal.).

5.2) SELECTION OF DESIGN TEMPERATURES AND HEAT EXCHANGER DESIGNS

Having specified the conceptual design of this combination hydronic heating

system, it is necessary to choose various design temperature levels and to size the various heat exchange components.

The nominal temperature rise of return air passing through a conventional air based heating unit is 44 to 47 C (80 to 85F). The design temperatures of the air entering and leaving the finned coil heating unit were chosen as 21.1C (70F) and 48.9C (120F) respectively. This relatively small air temperature rise helps to maintain a reasonable temperature difference between the two fluids (air and water) flowing through the finned coil heat exchanger. In an additional effort to maximize this air-water temperature difference the water heater set point design temperature was chosen as 71.1C (160F).

The choice of 71.1C (160F) water heater exit design temperature merits comment. At first glance, it would seem that the tank jacket losses could be wasteful since the nominal set point temperature of a residential water heater is about 48.9C (120F) to 60C (140F). However, as long as the combination heating and hot water unit is located within the heated space of the residence, these jacket losses eventually help to meet the comfort heating load and appear as energy used in a useful manner. Excessive jacket losses need not be a concern during the summer, non-heating, months since the furnace mode is not required and the water tank set point temperature can be reduced to the normal domestic hot water heater set point level. With this in mind, it can be seen that an even higher heater tank set point temperature during heating months would allow the use of less finned coil heat exchanger area (or would provide better heat exchange rates). It might also be noted that some dishwashers call for hot water at 65.6C (150F) or more which approaches the 71.1C (160F) design figure used for this study.

The design temperature of water leaving the finned coil heat exchanger was chosen as 37.8C (100F). Thus, the water temperature drop at the finned coil is 33.3C (60F). This temperature drop is somewhat greater than is called for by standard design methods. However, it is necessary that a relatively low finned coil exit temperature be achieved since it is this temperature that determines the temperature of the water in the water preheat tank. In turn, the temperature of the water in the preheat tank influences the temperature of the flue gases exiting the system. Thus, the overall system efficiency is affected.

During periods when the system operates as a furnace, it is clear that the lowest temperature to which flue gases can be cooled is 37.8C (100F). Referring to Figure 3.5, the maximum attainable efficiency under this operating condition is 94% (@ 35% excess air). However, it is apparent that a flue gas temperature of 37.8C (100F) is a lower limit and that a more realistic steady state efficiency of a well designed unit might be several percentage points below 94%.

On the other hand, when the unit operates as a domestic hot water heater the flue gas exit temperature could theoretically approach the mains water temperature. Assuming a mains water temperature of 10C (50F) it is seen that the efficiency of the unit could approach 97 or 98%. More realistically, the actual operating efficiency would again be several percentage points lower.

Considering the efficiency figures shown above and noting that the long term operating efficiency of the combination unit should fall somewhere in between the operating efficiencies associated with each of the two possible operating modes, it could reasonably be expected that the long term operating

efficiency of this unit should fall in the 90 to 95% range.

Having specified the temperatures of the air and water streams entering and exiting the finned coil unit, it is possible to size that heat exchanger. Based on a heat transfer rate of 22.3 kW (76,000 BTU/hr) at the heat exchanger (95% of the burner input) the flow rates of water and air are 9.6 liters/min. (2.11 GPM) and 2873 Kg/hr (6333 lbm/hr)(or 1487 CFM) respectively. Heat exchanger sizing was carried out using AEROFIN* bulletin number MP-72 for heating with water coils. The calculations carried out are included in Appendix C. These sizing calculations indicate that the required finned tube heat exchanger should consist of four tube banks connected such that the water flows in series through the four tube banks and in a counterflow arrangement with respect to the flow of air. Each of the tube banks consists of a face of ten finned tubes exposed to the air flow, each tube being 76.2 cm (30 inches) long. The ten tubes of any specific tube bank are to be connected in series. The total face area of heat exchanger exposed to air flow is 0.27 m^2 (2.93 ft^2).

The predicted pressure drops on the air and water sides of the heat exchanger are 0.84 cm H_2O (0.33 inches H_2O) and 4.6 m H_2O (15 ft H_2O) respectively which can be accomodated with readily available blowers and pumps. The pressure drop of the water passing through the heat exchanger represents a required pumping power of about 7.5 Watts (0.01 horsepower). This water flow could likely be achieved most simply using a small displacement pump.

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* AEROFIN Corporation (Canada) Ltd., Gananoque, Que.

In order to arrive at a suitable design of the water preheat tank, its heat exchange tubing was sized assuming a worst case of 37.8C (100F) water inlet temperature (space heating operation). Assuming that 15% of the burner input energy is given up in the preheat tank, 80% of the burner input energy is given up in the main water heater tank and that water exits the main tank at a temperature of 71.1C (160F), the steady state temperature of the water leaving the preheat tank can be shown to be 43.1C (109.5F). Entrance and exit flue gas design temperatures at the preheat tank were chosen as 232C (450F) and 43C (110F) respectively.

The temperature rise of the water passing through the preheat tank (5.3C or 9.5F) is quite small in relation to the temperature drop of the flue gas in the same tank (189C or 340F). Therefore, it was noted that the heat transfer in this case could be quantified using the standard definition of log mean temperature difference independent of the heat exchanger flow configuration. Using the design temperatures mentioned earlier, and shown in Figure 5.1, the log mean temperature difference present in the preheat tank is approximately 53C (95F).

Sizing calculations were carried out in order to arrive at a tube design for the preheat tank. These calculations are presented in Appendix D. These calculations assume that the controlling resistance to heat transfer between the flue gas inside the tubes and the preheat water is associated with the convective film coefficient on the flue gas side of the tube wall. The thermal resistances of the tube wall and of the water side convective film have been neglected. The design resulting from these calculations consists of seven tubes of 1.27 cm (½ inch) flow diameter, each running from the top to the bottom of the preheat

tank. Each tube is 4.57 m (15 ft) in length. Since each tube is quite long, it is necessary that each tube be formed into a helical shape in order that a reasonably small tank be used. These tubes must be installed such that each tube slopes downward at all points as it passes through the preheat tank so that the condensate can escape.

The pressure drop of the flue gas passing through the preheat tank is estimated to be 7.6 cm (3 inches) water. Clearly, the resistance to the flow of flue gas is dominated by the resistance associated with the preheat tank heat exchanger tubes. The rate of electrical energy used to move the flue gases is expected to be approximately 100 watts (0.13 horsepower).

5.3) PREHEAT TANK STORAGE CAPACITY

The storage capacity of the preheat tank is limited by the requirement that the temperature level of the preheat tank cannot be allowed to rise more rapidly than the temperature of the main tank during periods when the burner is on but no water is flowing. Such a period might occur immediately following a large draw of domestic hot water. Having met this requirement, the possibility of the preheat tank overheating before the main tank has reached its set point temperature is avoided. Assuming that 80% of the burner input energy is delivered to the main tank and 15% delivered to the preheat tank, it can be shown that it is necessary only to specify that the storage volume of the preheat tank be 19% or more of the storage volume of the main heater tank. For example, if the A.O. Smith, PGC-75, 284 liter (62.5 gallon) water heater was to be used, the preheat tank should have a capacity of at least 54 liters (12 gal.). Therefore, it can be seen that a preheat tank can be quite compact if the

layout of the heat exchange tubes is skillfully designed.

5.4) MATERIALS SELECTION

Since the hot water heater and finned coil heat exchanger are standard units the only materials selection required concerns the preheat tank and the various components contacted by the flue gas downstream of the preheat tank.

The preheat tank itself can be a simple steel vessel similar to a standard water heater tank. In the same manner, it should incorporate a sacrificial anode and have a reasonable level of insulation. The heat exchange tubes of the preheat tank must withstand temperatures in excess of 232C (450F) and must resist the corrosive action of the flue gas condensate. Thus, the tubes should be made of one of the high alloy steels or stainless steels identified in reference 27 as the most suitable materials for this application.

The flue gases are cooled sufficiently in the preheat tank such that any parts such as headers, ducts, blower wheels, etc. that contact the flue gas further downstream can be made of corrosion resistant, low service temperature, plastics.

5.5) COST CONSIDERATIONS

The cost to purchase this combination unit will be determined largely by the additional cost of a larger water heater instead of a standard unit, the cost of the finned coil unit and pump plus the cost of the preheat tank.

The price of the 23.4 kW (80,000 BTU/hr) input water heater is expected to be twice the price of a standard unit, or an additional $175 over the price of a standard water heater. The AEROFIN multi-pass finned coil heat exchanger described earlier can be purchased for $350*. The cost of this space heating system including the water heater, finned coil unit, blower, blower controls, pump, etc., but excluding the preheat tank and exhaust blower is potentially the same as the price of a conventional forced air gas fired furnace (assumed to be $700).

The preheat tank can be produced for about the price of a standard water heater ($175) and with the addition of the exhaust blower, condensate treatment and the extra controls needed to direct vent the flue gases, the estimated cost of the total system is $1000 to $1200.

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* Price quoted by Canada Blower and Forge, Kitchener, Ontario (October, 1981). The 4 row heat exchanger required can be constructed of a pair of 2 row heat exchangers which are priced at $175 (retail) each.

CHAPTER 6

THE AIR BASED HIGH EFFICIENCY FURNACE

6.1) THE CONCEPTUAL DESIGN

The great majority of residential space heating installations in Canada are air based units. Therefore, it was decided that a high efficiency air based furnace should be designed as part of this study.

As pointed out in Chapter 5, the cooling of combustion products is best carried out using two (or more) discrete heat exchangers each designed to perform in a specific manner. It was recognized in this case that the primary stage of heat exchange could readily be accomplished using the heat exchanger from a conventional furnace. Further heat exchange must be handled by a heat exchanger designed to withstand the corrosive nature of flue gas condensate.

It should be noted that the specific arrangement presented here is well suited to possible retro-fit application. Since the design is based on a two heat exchanger system, where the first consists of that which is found in a conventional furnace, an add-on package of blowers and heat exchanger equipment could readily be designed to upgrade the conventional furnace. Thus, the standard natural draft unit could be converted to a high efficiency, condensing, direct vent furnace.

The furnace design presented here is such that the flue gases are cooled to approximately 260C (500F) in an improved sectional heat exchanger and then are cooled further to about 37.8C (100F) or less in a plastic condensing heat

exchanger. Finally, the flue gases are cooled using a combustion air preheat, plastic, condensing, heat exchanger.

The plastic heat exchangers mentioned above are to be constructed of an extruded plastic sheet material called COROPLAST* which is made of a mixture of approximately 90% polyethylene and 10% polypropylene. By using Coroplast sheet the condensing heat exchangers can be built in a simple, compact and highly efficient manner. The use of plastic offers a resistance to possible condensate corrosion that few metals can duplicate.

The configuration of the Coroplast extruded sheet is similar to that of corrugated cardboard. Each sheet consists of two thin walls separated by a series of parallel ribs. The end view of this construction is shown in Figure 6.1. Nominal dimensions are also shown[30].

A cross-flow plastic heat exchanger can be constructed from Coroplast simply by stacking a series of sheets together such that the flow passages of each sheet run perpendicular to the flow passages of the two adjacent sheets. Once connecting ducts are added to four sides of the resulting block, a cross-flow heat exchanger is complete. This construction process is illustrated in Figure 6.2.

A heat exchanger constructed in this manner provides a relatively large

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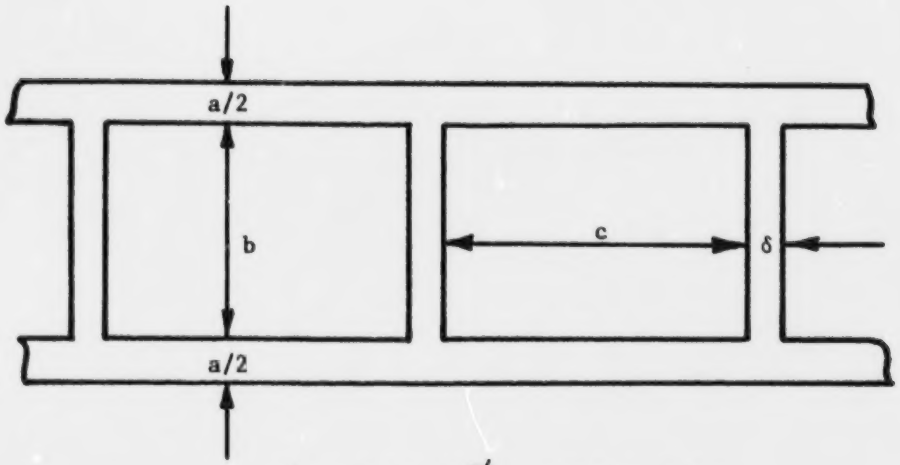
* Available from: Coroplast Division, Lorcon Inc.
927 Millwood Road
Toronto, Ontario M4G 1X4



$a/2 = 3.5 \times 10^{-4}$ m = 0.014 inches

$b = 3.63 \times 10^{-3}$ m = 0.143 inches

$c = 5.34 \times 10^{-3}$ m = 0.210 inches

$\delta = 3.0 \times 10^{-4}$ m = 0.012 inches

Figure 6.1: End View of Coroplast Extruded Sheet

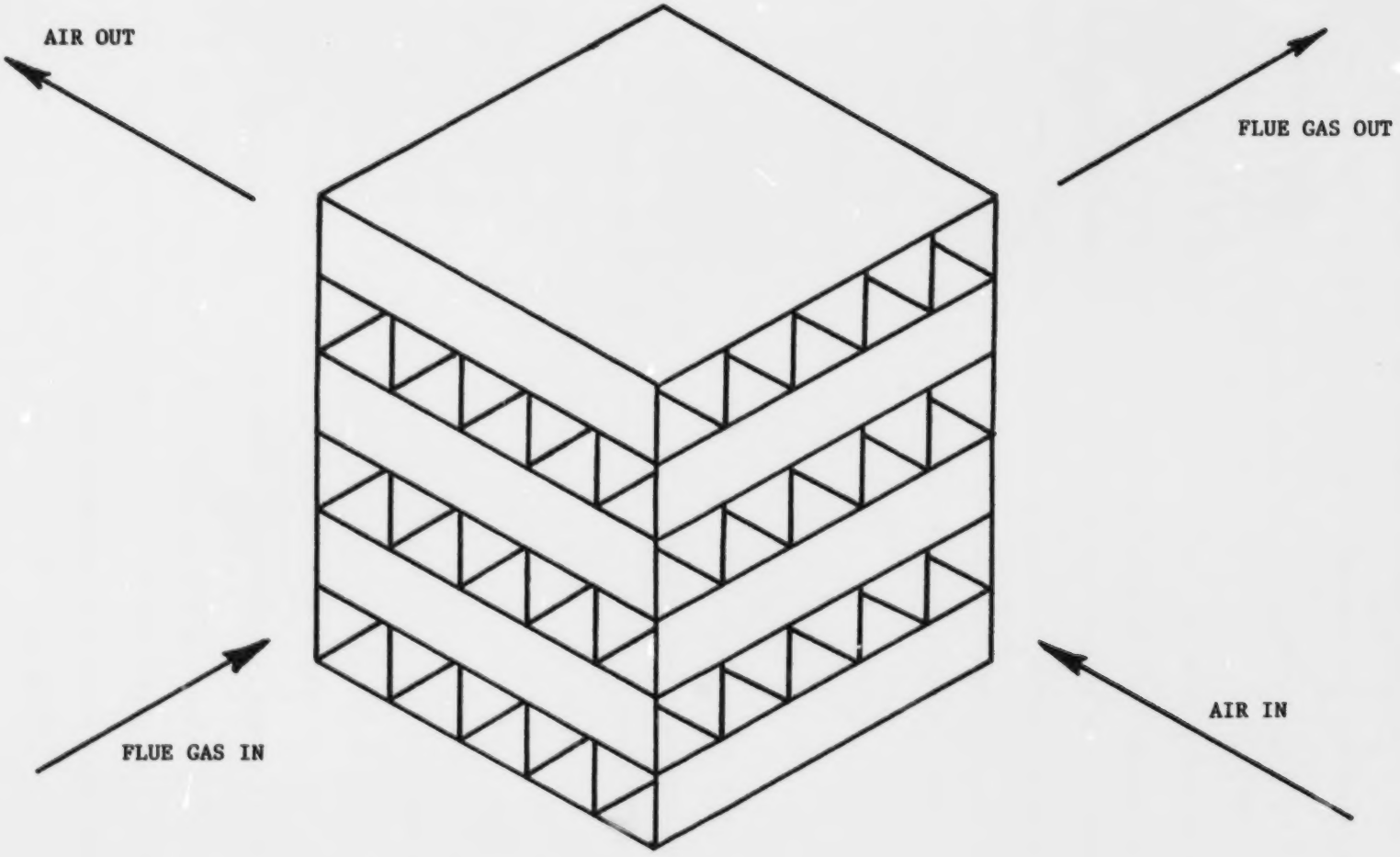


Figure 6.2: Construction of a Six Sheet Coroplast Heat Exchanger

heat exchanger area per unit heat exchanger volume. The ribs in the Coroplast sheet not only provide structural integrity but also act as fins to aid the heat transfer process.

Coroplast has a maximum service temperature (at which softening begins) of 121C (250F). Having noted that the temperature of the flue gas exiting the sectional heat exchanger is to be well above this maximum service temperature, a unique recirculation loop was designed in order to protect the plastic heat exchanger against overheating. The operation of this recirculation loop, as well as the other processes taking place, is illustrated in Figure 6.3.

6.2) RECIRCULATION FLOW CALCULATIONS

The purpose of the recirculation loop is to mix a sufficient amount of the cool saturated flue gas (which exits the condensing heat exchanger section of the furnace) with the relatively hot flue gas exiting the sectional heat exchanger in order that the temperature of the mixed flow entering the condensing heat exchanger does not exceed 121C (250F).

In order to calculate the required amount of flue gas recirculation, it is useful to use the molal flow rate of CO_2 to quantify the overall flow of constituents at various points in the furnace. The molal flow rate of CO_2 is denoted $\dot{n}_{CO_2,i}$ at the i^{th} state point. Performing an energy balance at the mixing junction upstream of the condensing heat exchanger, it can be shown that the ratio of the molal flow of CO_2 in the recirculation flow to the molal flow of CO_2 leaving the sectional heat exchanger is given by equation 6.1. $H(T_i)$ is the enthalpy of the flue gas at the i^{th} state point.

STATE 5: $CO_2 + 2H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2 + \frac{\delta_{CO_2,R}}{\delta_{CO_2,4}}(CO_2 + 2(1-f_7)H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2)$

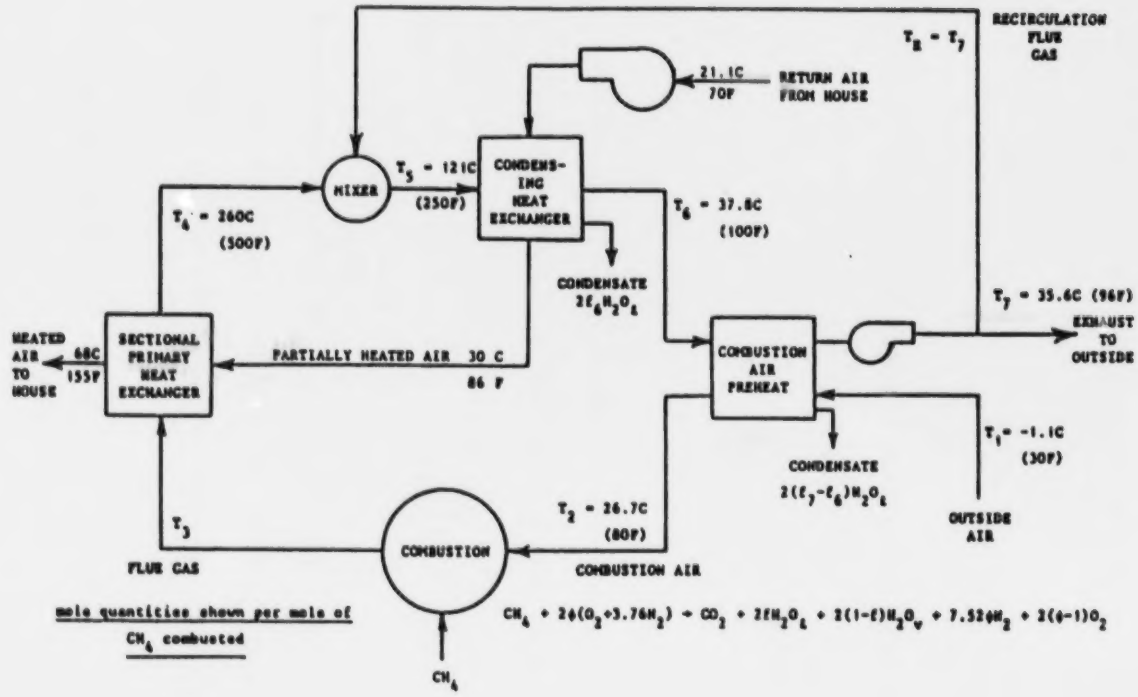


Figure 6.3: High Efficiency Furnace with Flue Gas Recirculation

$$\frac{\dot{n}_{CO_2,R}}{\dot{n}_{CO_2,R}} = \frac{H(T_4) - H(T_5)}{H(T_5) - H(T_R)} \tag{6.1}$$

where: $H(T_i) = (\hat{h}_{CO_2} + 2(1-f_i)\hat{h}_{H_2O,v} + 7.52\phi\,\hat{h}_{N_2} + 2(\phi-1)\hat{h}_{O_2})$ evaluated at $T = T_i$

The ratio of molal flow rates shown in equation 6.1 can be converted to a similar ratio of mass flow rates using equation 6.2.

$$\frac{\dot{m}_R}{\dot{m}_4} = \frac{\dot{n}_{CO_2,R}}{\dot{n}_{CO_2,4}} \cdot \frac{\hat{M}_R}{\hat{M}_4} \tag{6.2}$$

The molecular weight of the flue gas at the i^{th} state point, $\hat{M}_i$, is given as a function of the molecular weights of the various constituents as:

$$\hat{M}_i = \frac{\hat{M}_{CO_2} + 2(1-f_i)\hat{M}_{H_2O} + 7.52\phi\hat{M}_{N_2} + 2(\phi-1)\hat{M}_{O_2}}{1 + 2(1-f_i) + 7.52\phi + 2(\phi-1)} \tag{6.3}$$

Choosing design temperatures of T_5 = 121.1C (250F) and $T_7 = T_R$ = 35.6C (96F), equation 6.1 was solved as a function of the exit temperature of the primary heat exchanger, T_4, and the resulting curve is shown in Figure 6.4.

Using 35% excess air (ϕ = 1.35) and knowing f_7 = 0.64 (see Figure 3.2), equation 6.3 yields the result:

$$\frac{\hat{M}_R}{\hat{M}_4} = 1.04$$

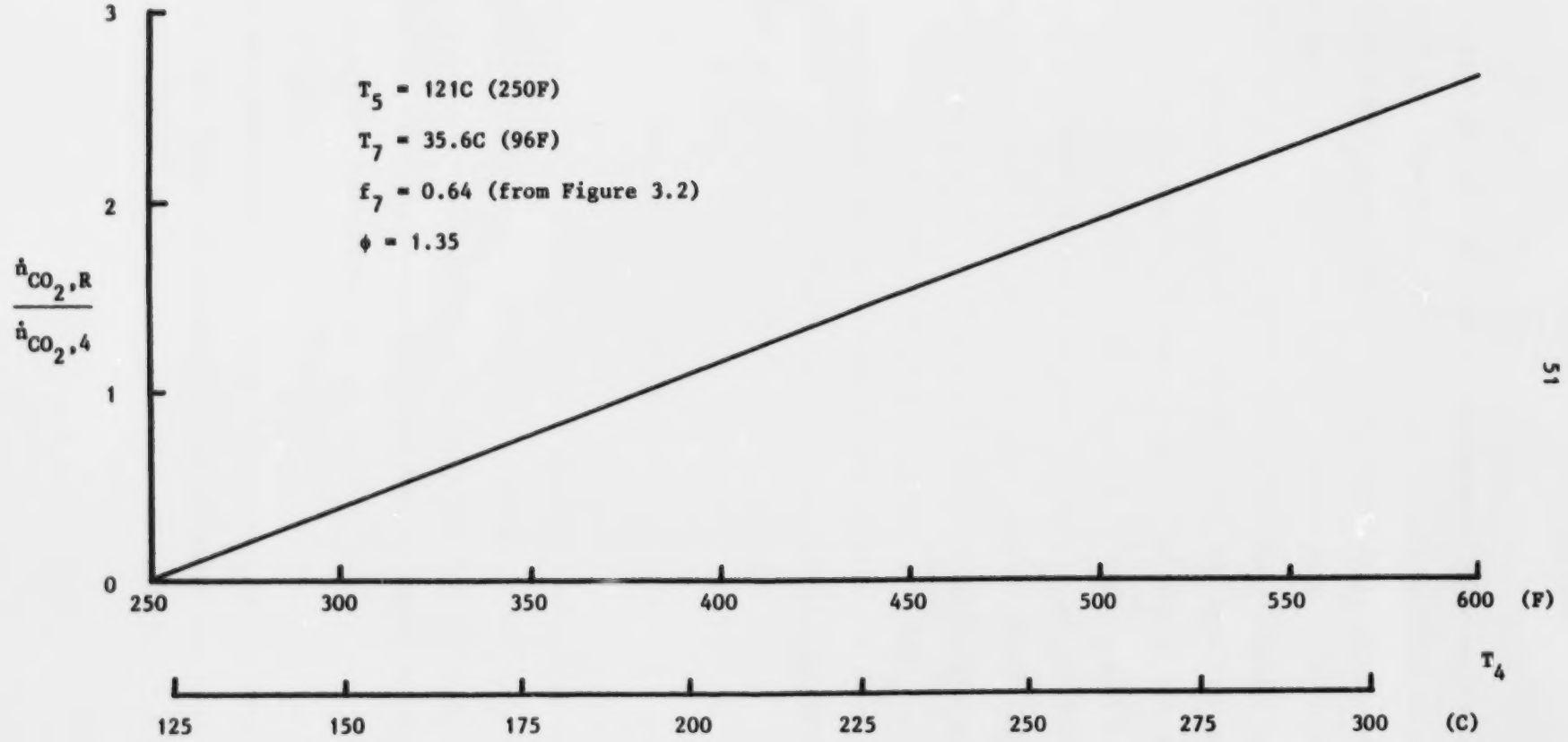


Figure 6.4: Flue Gas Recirculation as a Function of Primary Heat Exchanger Exit Temperature

Therefore, by equation 6.2, the ratio of recirculation gas mass flow to combustion product mass flow at state point 4 can be taken as being about 4% higher than the analogous mole ratio read from Figure 6.4. For example, if the flue gas exits the primary heat exchanger at T_4 = 260C (500F) then Figure 6.4 shows that (for 35% excess air) for each mole of CO_2 produced in combustion, about 1.9 moles of CO_2 are recirculated. Thus, the mass flow rate of recirculated flue gas is about 1.98 times the mass flow rate of flue gas leaving the primary heat exchanger. The mass flow rate of gas entering the condensing heat exchanger is about 2.98 times the rate at which gases leave the primary heat exchanger. It can also be noted that the volumetric flow rate of flue gas entering the condensing heat exchanger can be estimated as being approximately three times as great as the figure read from Figure 3.4. Therefore, the flow rate into the condensing heat exchanger is about 1.79 m^3/min (63 CFM)(@ T_5 = 121.1C (250F), 35% excess air and 23.4 kW (80,000 BTU/hr) burner input).

6.3) THE IMPROVED SECTIONAL HEAT EXCHANGER

As mentioned earlier, the air based high efficiency furnace presented here is to have a sectional type primary heat exchanger. Since the size of the downstream condensing heat exchanger is sensitive to the temperature at which the flue gas exits the primary heat exchanger, it seemed worthwhile to try and minimize this temperature.

Typical exit temperatures of existing furnaces fall in the range of 288C (550F) to 343C (650F). It was decided that a primary heat exchanger flue gas exit temperature of 260C (500F) is reasonable and could be achieved without

creating heat exchanger cold spots which are susceptible to corrosion. Further, it was reasoned that the additional heat exchange needed to drop the flue gas temperature to 260C (500F) could be achieved using standard sectional heat exchanger shells. In order to increase the heat exchange rate, several simple modifications are proposed which alter the flow patterns in and around the sectional heat exchanger shells. These changes are detailed subsequently.

In order to better understand the mechanisms by which heat transfer takes place in a typical sectional heat exchanger a simple heat transfer model was devised and the results produced using this model were compared to measured data taken from an actual sectional type furnace under operating conditions. The sectional shell was examined in two zones as shown in Figure 6.5. Knowing the temperature of the flue gas at the burner (adiabatic flame temperature) and the flue gas exit temperature, it was assumed that the flue gas temperature dropped by equal amounts in each of the two zones. (The heat transfer in the flue gas exit section of the exchanger was assumed to be small and was not modelled.) Similar approximations were made on the air side. Heat transfer coefficients and energy transfer rates were then calculated in each zone based on the mean flue gas and air temperatures. These calculations accounted for convective heat transfer on the air side and convective plus radiative heat transfer on the flue gas side of each zone. The energy transfer rates were then compared to the energy transferred as determined by the inlet and outlet flue gas and air temperatures. Using the energy transfer data produced from these calculations, the various temperatures of the two zones of heat exchanger were updated and a second set of calcuations was completed.

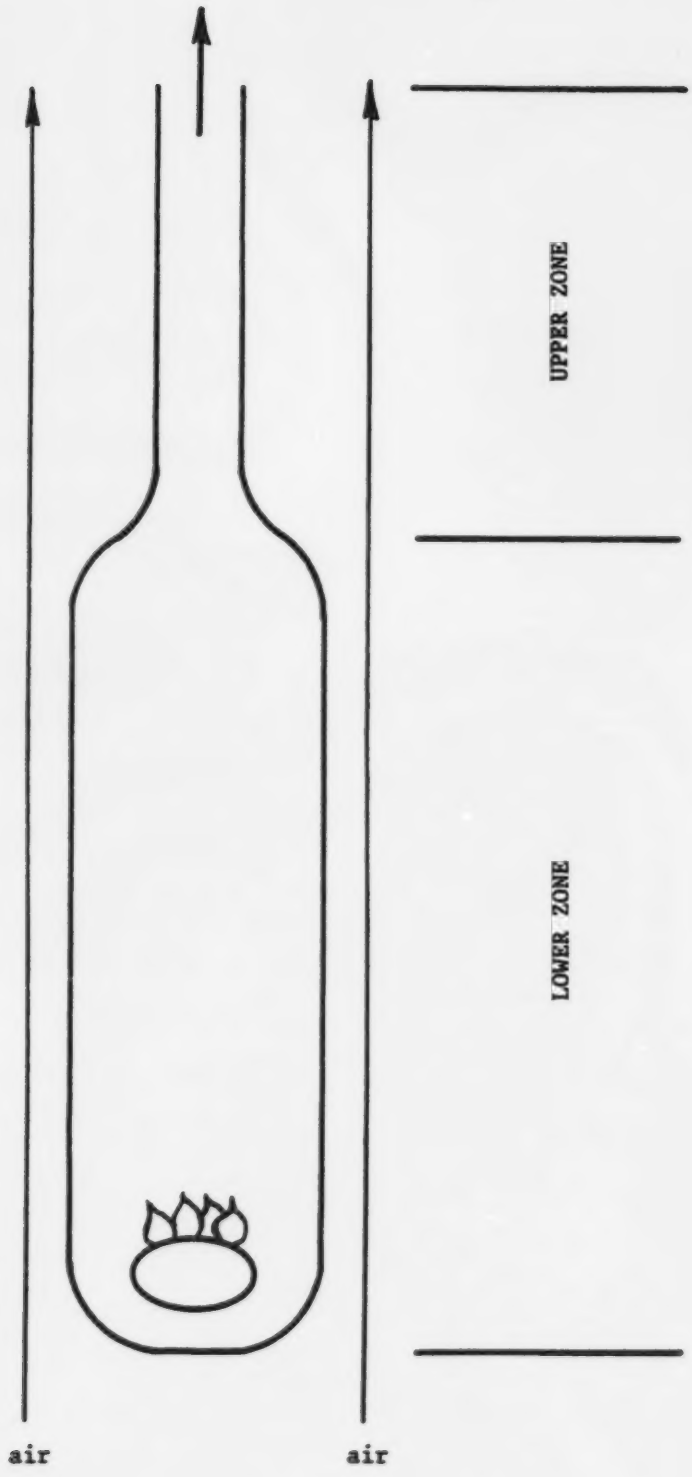


Figure 6.5: The Sectional Heat Exchanger Analysis Zones

When examining the measured heat transfer figures it was evident that the rate of energy transfer measured was quite close to the "rule of thumb" number* customarily used by furnace manufacturers. The first set of calculations carried out predicted heat transfer approximately 20% too high and the second set of calculations predicted heat transfer about 30% too low. It was noted that this range of agreement was quite reasonable and that subsequent estimates would be in closer agreement. Therefore, it was assumed that the mechanisms of heat transfer upon which the model was based must reasonably describe the actual mechanisms of heat transfer.

The more important pieces of data resulting from this analysis are presented in Table 6.1. From Table 6.1 the following observations can be made:

a) Heat transfer takes place predominantly by radiation in the lower zone of the heat exchanger. If the heat exchanger were to be redesigned with a small (shorter and/or narrower) lower zone, the radiative heat transfer would decrease dramatically and the convective heat transfer would increase only slightly (if the flow passage is made narrower). The larger the gas volume is, the "blacker" the gas appears to be. Therefore, if the lower zone is changed it should be increased, not reduced, in size.

b) A relatively small portion of the total energy transfer takes place in

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* Mr. E.C. (Ted) Clare pointed out the the average rate of heat transfer used for design purposes in a sectional heat txchanger is 1144 W/m^2 (42,000 $BTU/hr\ ft^2$).

TABLE 6.1

Sectional Heat Exchanger Heat Transfer Coefficients*

	Flue Gas Side Heat Transfer Coefficient (W/m^2C)		Air Side Heat Transfer Coefficient (W/m^2C)	Portion of Total Heat Transfer (%)
	(Radiative)	(Convective)	(Convective)	
First Set of Calculations				
Upper Zone	5.3	9.6	24.6	19
Lower Zone	25.5	5.1	44.8	81
Second Set of Calculations				
Upper Zone	2.7	1.2	24.6	15
Lower Zone	17.1	3.5	44.8	85

* effect of blackened flow diverters not included

the upper zone. Of the energy removed from the flue gas in the upper zone, most is removed by convective heat transfer. Radiative heat transfer contributes little because of the greatly reduced flue gas temperature and because of the shorter mean beam path length in this zone. It is also interesting to note that the heat transfer coefficient on the air side is considerably lower in the upper zone than in the lower zone because of the much wider air gap between the heat exchanger sections.

In view of these observations, several design changes are proposed.

a) Augment the heat transfer in the upper section flue gas side by introducing a solid blackened metal barrier in the flow path such that the flue gas must pass in between the barrier and the heat exchanger walls. This flow diversion increases heat transfer in two ways. First, the convective heat transfer coefficient between the flue gas and the heat exchanger walls will be increased because of the smaller hydraulic diameter of each of the two narrower flow passages. Second, the presence of the blackened flow diverter will couple the modes of heat transfer. Energy convected from the flue gas to the diverter can be transferred to the heat exchanger wall by means of the radiative mode. It has been estimated that this coupling of heat transfer modes alone can increase the combined (convective plus radiative) flue gas side heat transfer coefficient by approximately 40% or more if both the flow diverter and exchanger wall are blackened.

b) The heat transfer coefficient in the upper zone of the heat exchanger can be increased by introducing a similar blackened flow diverter in the

air side of the heat exchanger. This diverter should be constructed such that the thickness of the air flow passage is relatively uniform throughout the entire length of the heat exchanger.

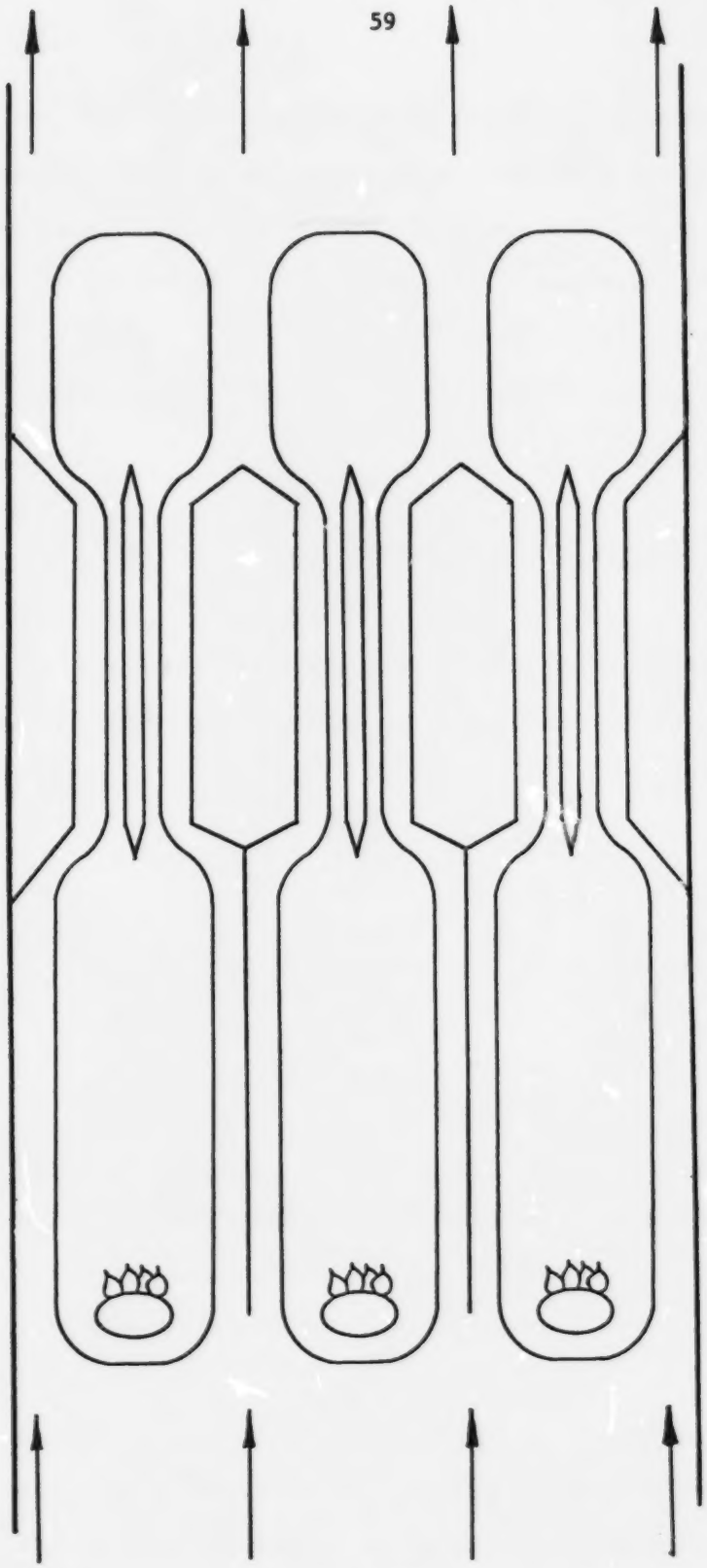
c) The heat transfer coefficient on the air side of the heat exchanger, in the lower zone, can be increased by introducing a blackened metal sheet midway between the adjacent heat exchanger sections. This metal sheet, or radiation curtain, does not divert air flow but can couple the heat transfer modes as described earlier. It has been estimated that this coupling of modes alone can increase the air side heat transfer coefficient by as much as 30%.

Figure 6.6 shows a sketch of a sectional heat exchanger as it would be seen with the improvements mentioned above.

6.4) SIZING THE CONDENSING PLASTIC HEAT EXCHANGER

As outlined earlier, the inlet and outlet flue gas design temperatures of the condensing heat exchanger are 260C (500F) and 37.8C (100F) respectively. Return air enters this heat exchanger at 21.1C (70F). Referring to Figure 3.4, the flue gas design temperatures indicate that 76% of the burner input energy is transferred in the primary heat exchanger and 18% of the input energy is released in the condensing heat exchanger. Therefore, if the total temperature rise of air passing through the furnace is 47C (85F) then the temperature rise of the air passing through the condensing heat exchanger is 9.0C (16.3F).

In order to size the condensing heat exchanger, it was analysed in two



AIR FLOW

Figure 6.6: Improved Sectional Heat Exchanger

sections. First, the section where the flue gas is above the dew point was sized and then the section where condensation takes place was sized. The dew point temperature was estimated by noting that the constituents (per mole CH_4 combusted) of the gas mixture entering the plastic heat exchanger are:

$$\text{molal flow at state point 5} = CO_2 + 2H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2 \qquad (6.4)$$

$$+ \frac{\dot{n}_{CO_2,R}}{\dot{n}_{CO_2,4}} \cdot (CO_2 + 2(1-f_7)H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2)$$

Using equation 6.4, it can be shown that the partial pressure of water vapour in the flow entering the condensing heat exchanger is 9.0 kPa (1.31 psia), where;

$\frac{\dot{n}_{CO_2,R}}{\dot{n}_{CO_2,4}} = 1.9$ (from Figure 6.4)

$f_7 = 0.64$ (from Figure 3.2)

$\phi = 1.35$

Referring to saturated steam tables, the dew point temperature is about 43C (110F). The design temperatures used to size the dry and wet portions of the condensing heat exchanger are shown in Figure 6.7.

Since the temperature change of the air passing through the dry portion of the heat exchanger is relatively small, it is valid to calculate heat transfer based on the log mean temperature difference (LMTD) even though the heat

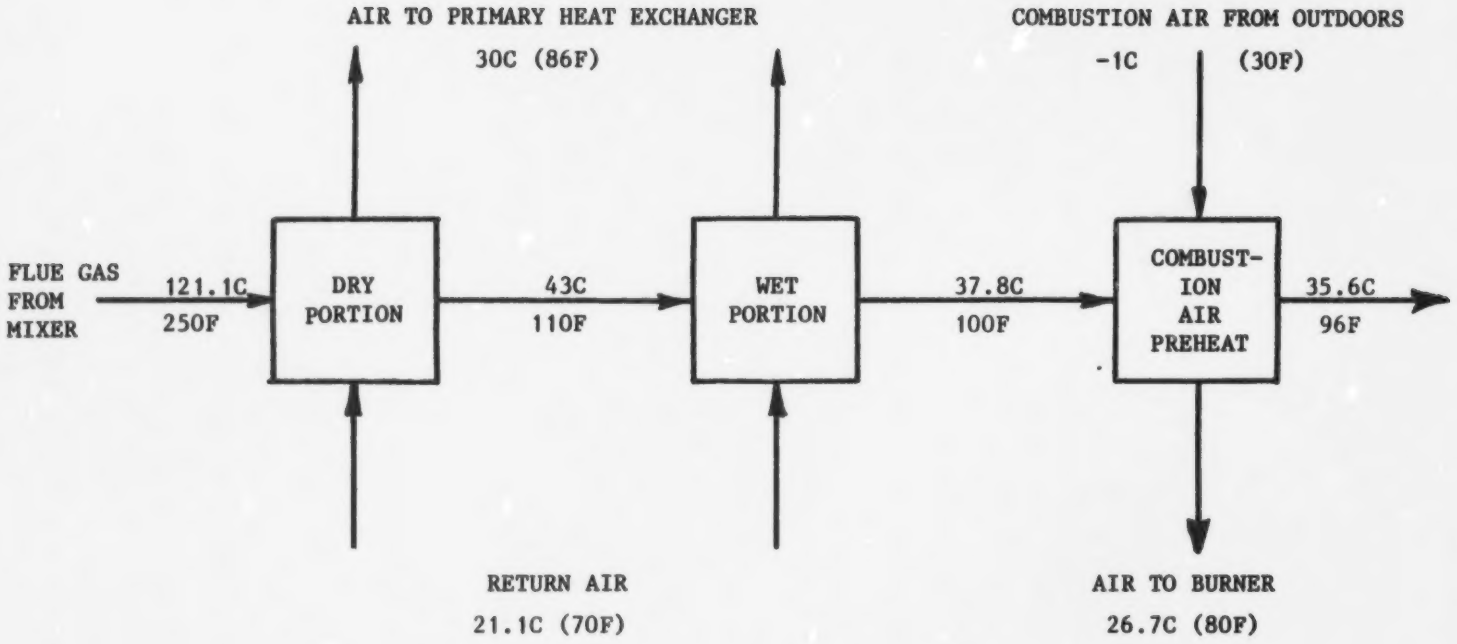


Figure 6.7: The Dry and Wet Portions of the Condensing Heat Exchanger

exchanger is not arranged in counterflow. The resulting LMTD is approximately 50C (89F). The rate of heat exchange in the dry section of the heat exchanger, $\dot{Q}_d$, can be expressed as:

$$\dot{Q}_d = A_{w,d} U_{w,d} (LMTD) \tag{6.6}$$

where; $A_{w,d}$ is the wall area of the dry portion of the heat exchanger.

$U_{w,d}$ is the heat transfer coefficient from flue gas to air based on wall area and dry heat transfer.

The heat transfer coefficient, $U_{w,d}$, was estimated using equation 6.7.

$$U_{w,d} = \left(\frac{1}{h_{FG}(1 + \eta_f A_f/A_w)} + \frac{1}{h_a(1 + \eta_f A_f/A_w)} \right)^{-1} \tag{6.7}$$

where; h_{FG} is the convective film heat transfer coefficient on the flue gas side

h_a is the convective film coefficient on the air side

A_f/A_w is the ratio of fin to wall area for Coroplast and is equal to 0.68

η_f is the fin efficiency of the Coroplast ribs and is equal to 0.24 (thermal conductivity of Coroplast taken as 0.15 W/mK).

It was assumed that the flow on both sides of the heat exchanger is fully

developed and laminar. The assumption of laminar flow can easily be shown to be true. The ducts between the Coroplast ribs have a width to height aspect ratio of 1.47. In this case the Nusselt number, Nu, for both flows is equal to 3.4[29]. The convective heat transfer coefficient, h, is expressed in terms of the Nusselt number as:

$$h = Nu \cdot \frac{k}{D} \tag{6.8}$$

where; k is the thermal conductivity of the fluid in question.

D is the hydraulic diameter of the flow passage ($D = 4.3 \times 10^{-3}$ m for Coroplast).

Making the approximation that the conductivity of both the flue gas and air is equal to the conductivity of dry air ($k = 0.028$ W/m K), the convective heat transfer coefficients in the dry section of the heat exchanger are:

$$h_{FG} = h_a = 22 \text{ W/m}^2\text{K} \qquad (3.88 \text{ BTU/hr ft}^2 \text{ F})$$

and $U_{w,d} = 12.7 \text{ W/m}^2\text{K}$ (2.1 BTU/hr ft² F)

The rate of heat transfer in the dry section of the heat exchanger was estimated using equation 6.9.

$$\dot{Q}_d = \dot{m}_5 C_{p,FG}(T_5 - T_6) \tag{6.9}$$

where; $\dot{m}_5 = \frac{\dot{m}_5}{\dot{m}_4} \cdot \dot{m}_4 \approx 3 \times 35.3 \text{ kg/hr} = 106 \text{ kg/hr}$

and $C_{p,FG}$ was assumed to be equal to the specific heat of air.

Solving equation 6.9:

$$\dot{Q}_d = 2.27 \text{ kW} \qquad (7760 \text{ BTU/hr})$$

Substituting this result into equation 6.6 yields:

$$A_d = 3.6 \text{ m}^2 = 39 \text{ ft}^2$$

Next, the size of the wet or condensing portion of the plastic heat exchanger was estimated. The heat transfer coefficient between condensing flue gas and air can be calculated as in equation 6.7 after having evaluated an effective wet film coefficient and a wet fin efficiency on the flue gas side of the heat exchanger.

For the purpose of estimating an effective wet film heat transfer coefficient on the flue gas side of the heat exchanger it was assumed that the saturated flue gas would behave in a manner similar to that of saturated air. Calculations were made according to the method presented in reference 31 for the analysis of condensing air cooling coils. In this analysis the effective wet side film coefficient, $h_{o,w}$, is given by:

$$h_{o,w} = \left(\frac{C_{p,a}}{b_w h_{c,o}} + \frac{y_w}{k_w}\right)^{-1} \qquad (6.9)$$

where; $C_{p,a}$ = specific heat of moist air

b_w = coefficient from Figure 12.18 of reference 31 evaluated at water film temperature

$h_{c,o}$ = convective film coefficient when no condensation takes place (22 W/m²K)

k_w = thermal conductivity of water

y_w = water film thickness

Considering a mean air temperature of 25.6C (78F) and a mean flue gas temperature of 42C (106F) the b_w coefficient of equation 6.9 was evaluated assuming a condensate film temperature of 37.8C (100F)(b_w = 1.8 BTU/lbm F). Solving equation 6.9 using nominal values of $C_{p,a}$, k_w and assuming y_w = 1.26×10^{-3} m (worst case) yields:

$h_{o,w}$ = 125 W/m²K (22 BTU/hr ft² F)

Calculating a wet side fin efficiency using $h_{o,w}$ instead of the dry film coefficient still yields a wet fin efficiency of 0.24. Solving for the overall wet heat transfer coefficient, $U_{w,w}$, using an equation similar to equation 6.7 yields:

$U_{w,w}$ = 21.7 W/m² K (3.83 BTU/hr ft² F)

In view of this result it was noted that the controlling thermal resistance in the wet section of the plastic heat exchanger is the air side

convective film resistance. Therefore, the wet section of the heat exchanger was sized neglecting the resistance to heat transfer on the flue gas side.

$$U_{w,w} = 25.5 \text{ W/m}^2\text{K} \qquad (4.5 \text{ BTU/hr ft}^2 \text{ F})$$

An estimate of the mean temperature difference between the flue gas and air sides of the heat exchanger was made as the difference between the mean air temperature of 25.6C (78F) and the mean flue gas temperature of 42.2C 106F). Thus, ΔT_m = 15.6C (28F).

Noting that 18% of the burner input energy is transferred in the plastic heat exchanger and that 9.7% of the burner input is transferred in the dry portion of the plastic heat exchanger, it follows that 8.3% of the burner input must be exchanged in the wet portion of the plastic heat exchanger ($\dot{Q}_w$ = 1.95 kW (6640 BTU/hr)). If the rate of heat transfer in the wet section can be expressed by equation 6.10,

$$\dot{Q}_w = A_w U_{w,w} (\Delta T_m) \qquad (6.10)$$

then it is possible to solve for the wall area of the wet portion of the heat exchanger, A_w, as;

$$A_w = 4.9\text{m}^2 = 53 \text{ ft}^2$$

The total heat exchanger wall area required is given by;

$$A = A_w + A_d = 8.5\text{m}^2 = 92 \text{ ft}^2 \qquad (6.11)$$

The volume of Coroplast heat exchanger needed to provide 8.5m² of heat exchanger wall area is 0.037m³ (1.3 ft³).

6.5) SIZING THE COMBUSTION AIR PREHEAT PLASTIC HEAT EXCHANGER

The combustion air preheat heat exchanger can be fabricated simply by extending the length (in the direction of flue gas flow) of the plastic heat exchanger examined in the previous section. In addition, duct connections must be made such that part of the face area seen on the air side of the heat exchanger accepts house return air and the remaining part, nearest the flue gas exhaust end, accepts combustion air.

In Chapter 3 it was shown that flue gas used to preheat combustion air by 27.8C (50F) would be cooled by approximately 6.1C (11F). However, the heat exchanger in question carries three times the flue gas flow as the one considered in Chapter 3. Therefore, the drop in flue gas temperature will only be about 2.2C (4F) and the preheat exchanger flue gas inlet and outlet design temperatures are 37.8C (100F) and 35.6C(96F) respectively. These temperatures are shown in Figures 6.7 and 6.3.

Since the flue gas temperature drop is relatively small, this heat exchanger can be sized using a log mean temperature difference (LMTD = 21.1C (38F)).

The rate of heat exchange in the preheat exchanger, $\dot{Q}_{ca}$, was calculated using the design air temperatures, giving $\dot{Q}_{ca}$ = 274 W (935 BTU/hr).

Using the same flue gas to air heat transfer coefficient as used in the previous wet heat transfer analysis, the required preheat exchanger heat transfer area is, A_{ca} = 0.5 m² (5.5 ft²).

6.6) DIMENSIONS OF THE PLASTIC HEAT EXCHANGER

Including the heat transfer area required for combustion air preheat, the total wall area required in the plastic heat exchanger is 9 m² (97.5 ft²). This corresponds to a Coroplast heat exchanger volume of 0.039 m³ (1.4 ft³). If the heat exchanger is constructed such that the length of the air flow passages is 8 cm (3.15 in) then the face area exposed to air flow would be 0.49 m² (5 ft²). If this face of the plastic exchanger is square it would measure 70 cm (27.5 in) along each edge.

The actual dimensions that should be used for this heat exchanger depend upon many factors including the size and shape of furnace in which it is to be used and the pressure drop that can be tolerated in each of the gas streams. Because of this, the choice of dimensions could more readily be considered during an experimental study or prototype development phase where the size and shape of the heat exchanger enclosure is known.

As stated in Chapter 4, blower power is not expected to be excessive. The flue gas pressure drop will be greater than that of a conventional furnace but the volume of gas to be moved is quite small.

6.7) PLASTIC HEAT EXCHANGER LEAKAGE

Inherent in the design of the Coroplast heat exchanger (as in any heat exchanger) is the possibility of leakage between the two gas streams. Although the various layers of Coroplast may be sealed carefully, it is wise to plan a design which allows for the possible occúrence of heat exchanger leaks.

To protect against leakage of the flue gas into the house air stream it is proposed that the furnace blowers be placed such that the house air is moved in a forced draft and the flue gas is moved in an induced draft fashion. With this blower configuration, any leak occurring in the plastic heat exchanger will result in leakage of air into the flue gas flow.

6.8) COST CONSIDERATIONS

The production of a condensing, direct vent, air based furnace entails two main costs in excess of the cost of a conventional furnace. These costs are associated with 1) the flue gas exhaust blower, the electronic ignition and control system and 2) the condensing heat exchanger.

The air based furnace presented in this chapter can be operated with exhaust blower, ignition and controls that are similar to those used by other direct vent furnaces currently available. The costs of these various components are expected to fall in the ranges of $50 to $75 for the exhaust blower, $100 to $125 for the electronic ignition and $100 to $125 for additional controls.

The cost of the Coroplast material used to produce the plastic condensing heat exchanger is approximately $15 per 2.97 m^2 (32 ft^2) sheet. Therefore, the materials cost of the plastic to build the condensing heat exchanger sized earlier in this chapter will be approximately $45. The cost of labour to build this heat exchanger is not known but it is of interest to note that the Coroplast can be processed with light duty machinery and no heavy or expensive tooling should be required.

In view of the costs mentioned above, the recirculation type, high efficiency furnace presented here should possibly be produced for a cost to the consumer of $300 to $400 above the cost of a conventional furnace. Clearly, this furnace design exhibits a great deal of performance and economic potential.

CONCLUSIONS

As a result of this study two high efficiency, condensing, direct vent heating unit designs have been developed. One design is an air based furnace and the other is capable of providing domestic hot water as well as space heating. Preliminary cost estimates indicate that both of these designs should be very competitive when compared to equipment currently available.

In addition to the potential that these designs possess as complete heating units, the air based, recirculation loop furnace design also has excellent potential as a retro-fit unit.

It is felt that the recirculation loop process incorporated in the air based furnace is a patentable feature.

The redesign of a conventional furnace to achieve high efficiency through condensation of flue products does not require a breakthrough in technology but utilizes state of the art methods. In essence, it involves additional heat exchanger equipment, a power vented exhaust and minor alterations to the controls to maintain safety and reliability. As a result, the excellent standard of service and low maintenance requirements characteristic of conventional furnaces should also be achievable with the designs presented here. All of the system components (blowers, controls, igniters, burners, etc.) are readily available and have been proven.

It should be noted that heat transfer predictions characteristica'ly are accompanied by a large margin of error. A discrepancy of 20% between predicted and measured heat transfer data is considered normal while larger

discrepancies are not unusual. Therefore, the calculated heat exchanger data presented in this report should be viewed as a starting point for experimental refinement of the heating system designs.

It is the opinion of the authors that neither of the designs presented in this study interfere with any patents found during the patent literature review.

RECOMMENDATIONS

Both of the heating system designs presented in this report have a great deal of potential. Since these two designs are vastly different, it is recommended that the development of both designs should be further pursued.

The two designs presented are at the development level where little additional theoretical work can be carried out. Therefore, it is recommended that both of these designs be refined in an experimental manner. It is also recommended that prototypes of both of these designs be produced and tested.

In view of the fact that a large number of conventional gas fired furnaces are currently in use, it is recommended that a retro-fit package based on the recirculation loop - plastic heat exchanger be designed. It is also recommended that this retro-fit system be refined experimentally to the point where prototypes can be produced and tested.

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APPENDIX A

PATENT LITERATURE SUMMARIES

A.1) THE PATENT SEARCH

The patent literature searches disclosed ninety-four patents of various degrees of relevance to the topic in question. The majority of these patents have been classified under one of the following subsystem headings.

a) Air based heating systems

b) Combined air/water systems

c) Exhaust flue heat recovery devices

d) Heat pump and/or heat pipe systems

e) Heat exchangers

f) Microwave and biomass furnaces

A list of the various U.S. Patent numbers, as they are grouped in each of these subsystem categories, is shown in Table A.1. A summary of the patents found, according to these subsystem groupings, is presented in the following sections. Having reviewed the patent abstracts and/or titles, copies of the complete patents corresponding to the most interesting abstracts were obtained. These patents are included in Appendix B.

TABLE A.1

HEATING SYSTEM PATENT LITERATURE SEARCH U.S. PATENT NUMBERS

AIR	AIR/WATER	EXHAUST FLUE HEAT RECOVERY DEVICES	HEAT PIPE/ PUMP	HEAT EXCHANGERS	MICROWAVE/ BIOMASS*
4256081	4257557	4216903	4171088	4274581	4217878*
4217877	4158438	4206875	4136733	4241874	3891817
4217876	4158386	4176787	4124177	4203415	
4215669	4147301	4175699	3993244	4171772	
4202318	4106692	4175698	3884292	4171089	
4175538	4090492	4171773	3880230	4160440	
4164210	4069807	4158439		3916869	
4138062	4065054	4147303		3908629	
4106474	3934554	4139152		3610333	
4079888	3833170	4136731		3610331	
4078541	3763849	4134545			
4047515	3763848	4124178			
4044950	3567905	4066210			
4033320	3563225	4050628			
4020822	3443062	4050627			
4006728	3236226	4037786			
3999710		4018381			
3982525		3999709			
3926173		3934798			
3916991		3930489			
3906925		3896992			
3871354		3404674			
3504850					
3444854					
3200810					
3007467					

A.2) AIR BASED HEATING SYSTEMS

With a few exceptions, the furnaces described in the literature are "conventional" in design (i.e. solid, liquid or gaseous fuel based, natural draft and fitted with atmospheric burners).

A number of patents deal with multiple fuel utilization incorporated in a single unit and/or one or more heat exchangers either for direct heat exchange or fabricated from refractory material for thermal storage and off cycle heat release.

One patent (P.N. U.S. 4078541) describes a multistage liquid/solid fuel burning furnace with increased combustion efficiency and heat transfer characteristics due to the introduction of turbulence in the combustion chamber.

A hot air heater with induced draft is described in P.N. U.S. 4069807. The combustion products are cooled (to a temperature between 170F and 190F) and humidified by being passed through a water bath and then are passed through a thermoplastic heat exchanger where the water vapour is recondensed. This water can then be returned to the water bath of the initial heat exchange process. The plastic heat exchanger is specified as having 3 to 10 square feet of air surface area per 1000 BTU/hr of burner input capacity.

A pulse combustion system (P.N. U.S. 4164210) for the heating of air describes a furnace system wherein the combustion gases are cooled to a temperature below their dew point. The secondary heat exchanger is fabricated from a high thermal conductivity material which also is corrosion resistant and enhances the muffling action to reduce noise output.

In addition to the systems mentioned above, two patents deal with sealed combustion furnaces (P.N. U.S. 3871353 and P.N. U.S. 3444854) which are conventional in design.

A.3) COMBINED AIR/WATER SYSTEMS

The residential heating systems of this type that were found contained standard combustion chambers, were liquid or gas fueled, had liquid and/or air heat exchangers to cool the combustion products where one or more heat exchangers were connected in series. Heat delivery for space heating was generally accomplished by means of a forced air circulating system. In some designs the liquid heat exchange section of the furnace was also used for cooling purposes.

This sub-system category of air/water heating systems includes boilers, domestic hot water heaters with liquid to forced air heat exchangers and combination multi-stage combustion chamber and heat exchange facilities. However, no "novel" designs were encountered.

A.4) EXHAUST FLUE HEAT RECOVERY DEVICES

The flue stack heat recovery devices patented are water and/or air based heat exchange systems, located either in the chimney or in the flue pipe upstream of the chimney.

Water based recovery devices are connected in series with domestic hot water heaters and/or liquid to forced air heat exchangers. An appropriate pump and control system are included.

Air based recovery devices are flue products to air heat exchangers which include blower attachments. One such device utilizes a porous medium through which air is forced.

A.5) HEAT PUMP AND HEAT PIPE SYSTEMS

In the literature searched, two heat pump and four heat pipe heating systems were described.

Heat removal in the case of the heat pipes was by means of convection or forced air circulation.

One patent (P.N. U.S. 4124177) describes a three stage heat pump for heating purposes.

A.6) HEAT EXCHANGERS

The heat exchangers patented offered no "novel" concepts.

However, one heat exchanger (P.N. U.S. 4171089) is of some relevance to the design of a high efficiency furnace. This is a heat exchanger with vertically oriented tubes through which the flow of flue products is induced. This exchanger communicates with combustion air drawn from outdoors, house return air and the combustion products in such a way that the flue products are cooled to a condensing state.

A.7) MICROWAVE AND BIOMASS FURNACES

The biomass (crop residue) furnace patented is a large output heater device intended for agricultural use.

The hydronic heating system (P.N. U.S. 3891817) is comprised of a microwave heater, storage tank and heat exchanger with flow and input energy controls.

APPENDIX B

PATENTS

United States Patent [19]

Hartig

[11] **4,069,807**

[45] **Jan. 24, 1978**

[54] **HOT AIR HEATER**

[75] Inventor: **Martval John Hartig, Wilmington, Del.**

[73] Assignee: **E. I. Du Pont de Nemours and Company, Wilmington, Del.**

[21] Appl. No.: **675,765**

[22] Filed: **Apr. 12, 1976**

[51] Int. Cl.² **F24H 3/06**

[52] U.S. Cl. ..... **126/110 R; 60/39.57; 126/360 A; 165/166**

[58] Field of Search ..... 126/360 R, 360 A, 113, 126/110 R, 110 B; 60/39.57; 165/DIG. 8, 166, 169; 237/9 B

[56] **References Cited**

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3,568,658	3/1971	Brock	126/360 A
3,733,244	5/1973	Sanberger et al.	165/166
3,940,301	2/1976	Straw et al.	165/166 X

Primary Examiner—William E. Wayner
Assistant Examiner—William E. Tapolcai, Jr.

[57] **ABSTRACT**

A high efficiency hot air heater is disclosed in which the combustion products of a flame are forced through water in a contacting device. The combustion products and whatever moisture is evaporated are passed through a heat exchanger to heat air such as for a home. The moisture which condenses in the heat exchanger is returned to the water contactor and the combustion products are vented to the atmosphere. Due to the low temperature of the combustion products and evaporated water, a plastic heat exchanger can be used, which is low enough in cost to permit using a large unit for maximum heat utilization.

7 Claims, 3 Drawing Figures

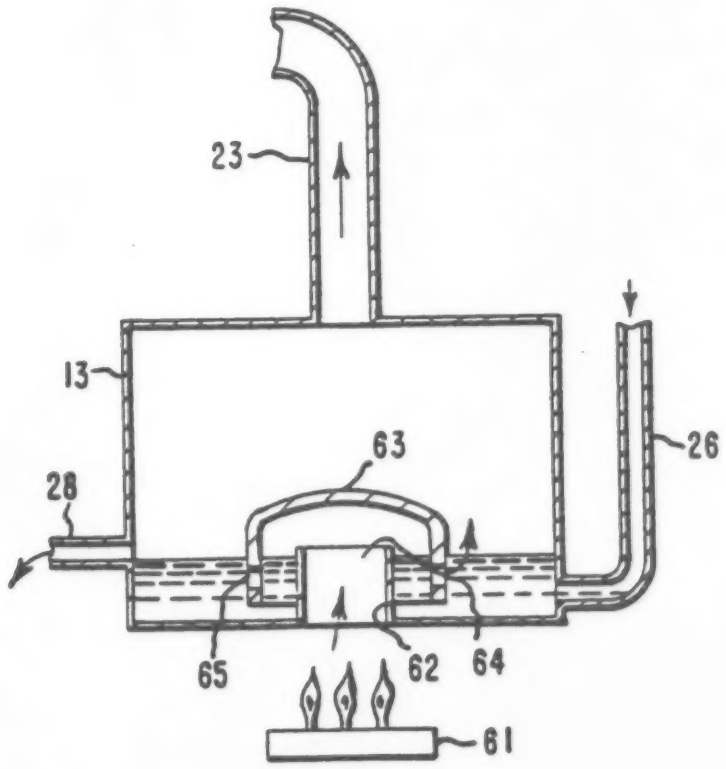


FIG. 1

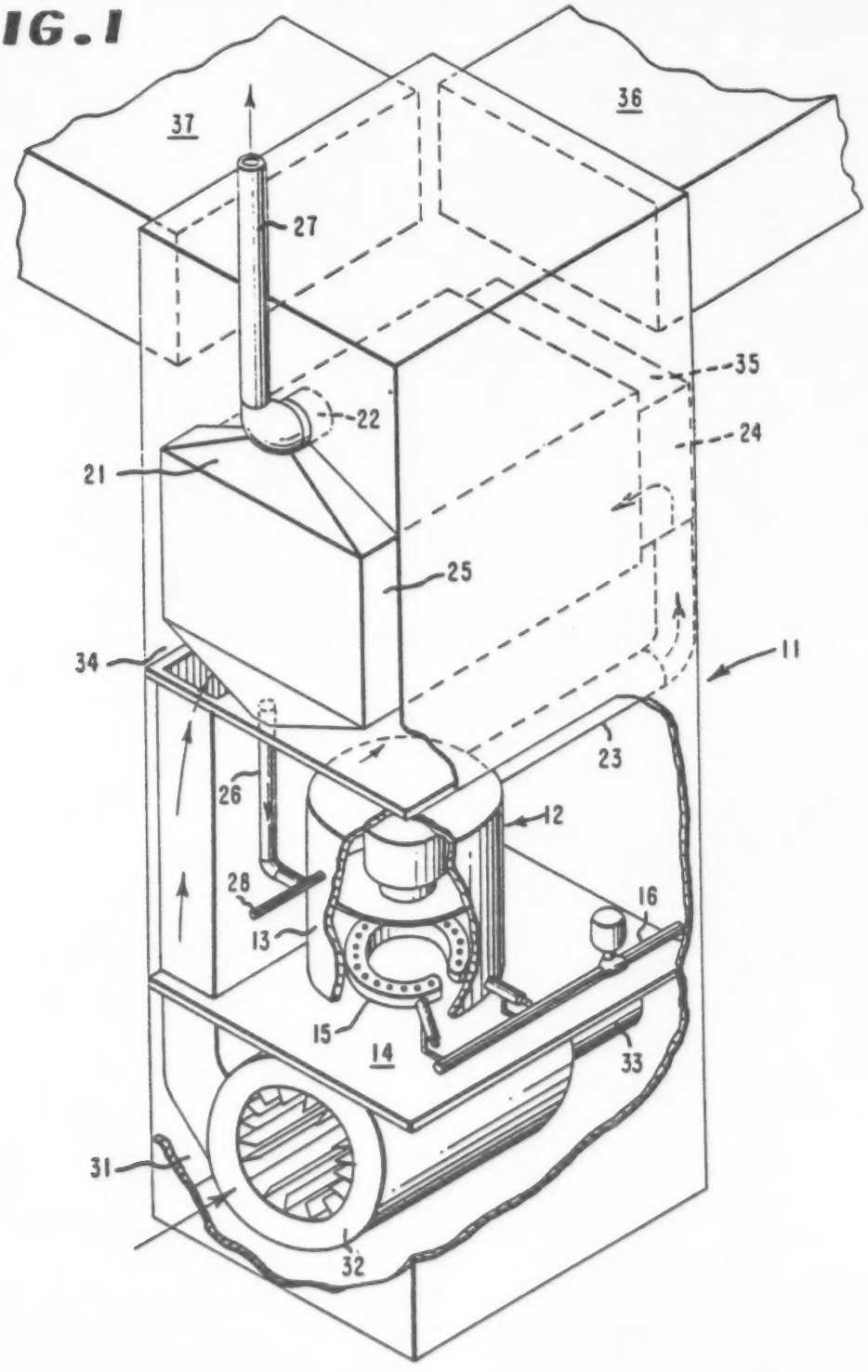


FIG. 2

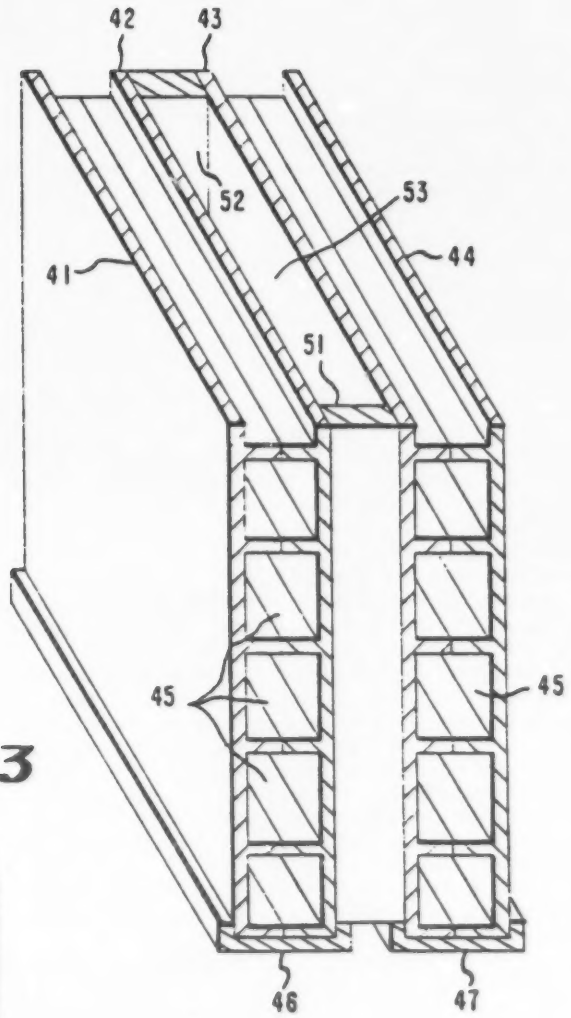
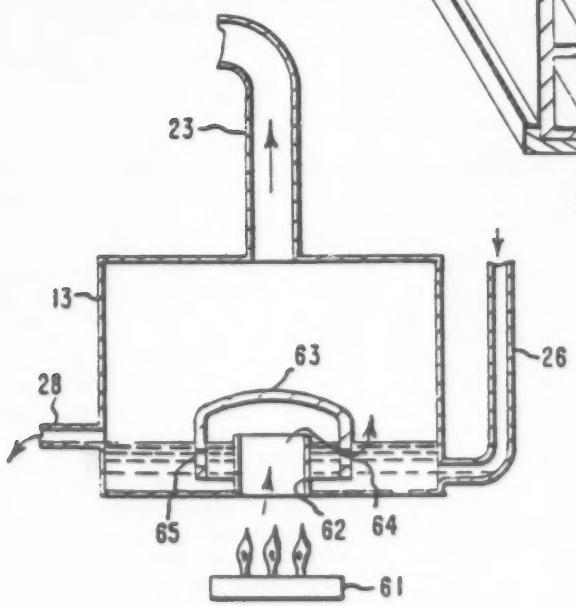


FIG. 3



HOT AIR HEATER

BACKGROUND OF THE INVENTION

At present most hot air heaters burn a fuel which generally is either No. 2 fuel oil or gas, the combustion products of which pass through a heat exchanger and then are vented into the atmosphere. The air from the building or other space to be heated is passed from the cold air return through the other side of the heat exchanger and back through the building. Due to the high temperature of the gas going up the vent to the atmosphere a considerable amount of heat is lost. Generally conventional gas fired heaters are about 70 to 75% efficient while conventional oil fired heaters are about 65 to 70% efficient. The present heater generally will provide efficiencies of up to about 98% which enables a considerable reduction in the amount of fuel consumed in heating the building or other space to be heated.

SUMMARY OF THE INVENTION

The present invention relates to a hot air heater having improved efficiency. While any combustible fuel can be used by the heater, the conventional gas or No. 2 fuel oil are the preferred fuels. The combustion products from the burning fuel and some excess air are drawn or forced through a water contacting device to cool the combustion gases and transfer heat to the water, part of which is thereby evaporated. The combustion gases and evaporated water are then transported through a heat exchanger where they are cooled and most of the evaporated water condensed and returned to the water contacting device. The combustion gases and uncondensed water are vented to the atmosphere using a forced air system. Air to be heated for a home, building or other use is passed through the other side of the heat exchanger to receive heat from the moist combustion gas. Combustion products' sensible heat which is transferred to evaporating water as latent heat is recovered by condensation in the room air heat exchanger, which also cools the combustion products to be vented minimizing heat loss up the stack.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an overall view of one embodiment of the heater of the present invention.

FIG. 2 is a detailed blow-up of the interior of a section of one embodiment of the heat exchanger used in the present invention.

FIG. 3 is a cross-section of the water contacting bubble cap assembly shown in FIG. 1.

DETAILED DESCRIPTION

Referring now to the illustrative example embodiment in FIG. 1, a home heater indicated generally as 11 is fitted with a burner and water contacting device which here is a bubble cap assembly and housing indicated generally as 12. The housing 13 is mounted on shelf 14. The bottom of housing 13 contains burner manifold 15 which is fed with fuel by pipe 16.

The top of manifold 21 is fitted with blower 22 which serves to draw combustion products and evaporated moisture through bubble cap assembly and housing 12 and draw such combustion products and evaporated moisture through duct 23 into heat exchanger manifold 24 through heat exchanger 25 into manifold 21. Most of the evaporated moisture condenses in the heat exchanger 25 so that manifold 21 is adapted to collect moisture and return it to bubble cap assembly and housing 12 by means of pipe 26. The combustion products are then vented to the atmosphere by means of stack 27. Blower 22 forces combustion products up the stack, said force replacing the thermal draft for a conventional furnace. Housing 13 is fitted with drain 28 to remove water which is formed by combustion of fuel and condensed in the heat exchanger 25. Pipe 26 can be used to occasionally fill the bubble cap.

Another view of the bubble cap embodiment is shown in FIG. 3 which is an elevation cross section of that part of the heater. Burner 61 produces hot combustion products which are passed up short chimney pipe 62 into bubble cap 63 and exit the bubble cap under the water through openings around the skirt at 64 and 65. The bubble cap is a cylinder closed at the top but open at the bottom as shown. Combustion products bubble through and evaporate some of the water in a pool at the bottom of housing 13. The top of chimney 62 is high enough to keep water from overflowing onto the burner, the water level being determined by placement of overflow outlet 28. Moisture laden combustion products now cool enough for a plastic heat exchanger pass from housing 13 through duct 23 to the room air heat exchanger.

Returning now to FIG. 1, a conventional home cold air return not shown is connected to plenum chamber 31 containing blower 32 driven by motor 33 and adapted to force cold home air into plenum chamber 34 through heat exchanger 25 and thence into plenum chamber 35 which is connected to hot air ducts 36 and 37 which feed hot air to the home. The controls used to prevent plenum chamber overheating, for burner ignition, and for blower operation are not shown because they are conventional. If desired, a coil of pipe can be placed in contact with the water in the bubble cap housing and connected to the domestic water heater to provide heating capacity therefor.

Referring now to FIG. 2 a partial section of one embodiment of the heat exchanger assembly is shown. This heat exchanger is formed from ribbed sheets 41, 42, 43 and 44 which are assembled with the horizontal ribs of one sheet contacting the ribs of another ribbed sheet so as to form ducts 45 through which the combustion products can pass. The ends of each pair of ribbed sheets are held together by channels 46 and 47. The pairs of ribbed sheets 41, 42, 43 and 44 are spaced apart from each other by means of vertical bars 51 and 52 to form opening 53 through which the home air passes upward. It is to be understood that there normally will be a plurality of bars similar to bars 51 and 52 in the heat exchanger and generally the spacing will be about one bar every 2 to 6 inches. Generally these bars will be about 0.01 to 0.03 inches wide by about 0.05 to 0.25 inches high to provide openings 53 corresponding to the height of the bars. The channels 46 are preferably heat sealed to ends of the ribbed sheets. The individual ribs of the ribbed sheet need not be fastened to the ribs of the opposite ribbed sheet. If desired, the ribs of one ribbed sheet can be formed with tongues and the ribs of the other ribbed sheet with grooves adapted to receive and grip the tongues in place. Generally the ribs will be of a height and spacing to form openings 45 which are from 0.05 to 0.2 inches wide and from 0.02 to 0.15 inches high. These dimensions have been found to achieve good heat transfer with low pressure drop across the heat exchanger while still being readily achievable using conventional construction techniques.

Generally the heat exchanger will have from 3 to 10 square feet of house air surface per 1,000 Btu per hour burner capacity. A conventional 4 inch bubble cap common in chemical distillation equipment is adequate for a heater of 100,000 Btu/hour capacity. The bubble cap and housing and the water level therein is usually sized to provide combustion products which are fed to the heat exchanger at from 175° to 190° F. While a single stage heat exchanger is shown it is to be understood that multiple stage heat exchangers can be used if desired.

Design trade-offs including use of alternate materials of construction may call for a higher temperature so long as it is safely below the limit for the heat exchanger material. Other design alternatives may be used such as types, shapes, and arrangements of components, spacing and fastening means, and types of control devices; all known in the art and within the spirit of the invention. If desired, for example, the bubble cap can be replaced by other water contacting means such as drawing the combustion gas through a falling curtain of water or by drawing the combustion gas down a passage and through a pool of water at the bottom of the passage. Alternatively, a water spray chamber, wet mats or wet plates can be used. Obviously many other modifications of the basic invention can be made. For instance, a salt can be added to the water in the bubble cap to raise the temperature of the moist combustion products. One advantage of the system of the present invention is that a relatively small burner and bubble cap can be placed on the floor in a small space and the heat exchanger, which heats the building air can be placed in existing ductwork thereby minimizing the space taken up by the heater.

The ribbed sheets and preferably also the spacer bars and end channels in the specific example cited are formed by extrusion of a synthetic thermoplastic resin from a suitable die. The ribbed sheets can conveniently be formed by the technique shown in any of U.S. Pat. Nos. 3,509,005; 3,515,778; 3,635,631; or 3,664,906. The thermoplastic resin used should have a softening point about 220° F or above and be reasonably rigid. A modulus of elasticity in flex of greater than 10,000 psi provides adequate rigidity. The preferred thermoplastic resins for use in forming the ribbed sheets are high density polyethylene, isotactic polypropylene, acrylonitrile-butadiene-styrene resins, styrene-acrylonitrile copolymers, and fluorocarbon resins such as fluorinated ethylene-propylene copolymers and polytetrafluoroethylene. The high density polyethylene and isotactic polypropylene are especially preferred because of their low cost. The invention, however, is not limited as to materials of construction. For example, anodized aluminum foil or copper-plated steel heat exchangers can be used if desired.

I claim:

1. A heater comprising a burner, means to conduct combustion products into contact with a liquid comprising water, said means being sized to provide combustion products at a temperature of from about 175° F to about 190° F, a blower adapted to transport from said burner through said liquid and urge said combustion products through a heat exchanger, which is made from a thermoplastic resin, which has from 3 to 10 square feet of air surface per 1,000 Btu per hour burner capacity, which heat exchanger is adapted to exchange the heat from said combustion products with forced air.

2. A heater comprising a burner, a chimney adjacent said burner which chimney protrudes through a body of water into a bubble cap, means to maintain said water at a predetermined level such that the combustion products from said burner are at a temperature of from about 175° F to about 190° F, a blower adapted to transport combustion products from said burner through said water and urge said combustion products through a heat exchanger, which is made from a thermoplastic resin, which has from 3 to 10 square feet of air surface per 1,000 Btu per hour of burner capacity is adapted to exchange heat from said combustion products with forced air.

3. The heater of claim 2 wherein means are provided to remove condensate from said heat exchanger and return said condensate to said bubble cap.

4. The heater of claim 3 wherein there are provided a plurality of thermoplastic resin passages adapted to conduct air, and there are provided adjacent thereto passages adapted to conduct combustion products in a direction essential at right angles to the direction in which the air is conducted.

5. A process of heating air comprising burning a fuel to produce combustion products, passing the combustion products into contact with a liquid comprising water to produce a gas mixture of moisture and combustion products having a temperature of from about 175° F to about 190° F and passing said gas mixture through a heat exchanger made of a thermoplastic resin to heat air.

6. The process of claim 5 wherein moisture is condensed in the heat exchanger and returned to liquid comprising water through which is contacted with the combustion products.

7. The process of claim 5 wherein the heat exchanger is operated in a plurality of stages with the flow of air at essential right angles to the flow of moisture and combustion products.

* * * * *

United States Patent [19]

Hollowell

[11] **4,164,210**

[45] **Aug. 14, 1979**

[54] **PULSE COMBUSTION SYSTEM FOR HEATING OF AIR**

[75] Inventor: **George T. Hollowell, Mayfields Heights, Ohio**

[73] Assignee: **Gas Research Institute, Chicago, Ill.**

[21] Appl. No.: **828,996**

[22] Filed: **Aug. 30, 1977**

[51] **Int. Cl.**[2] **F24H 3/08**

[52] **U.S. Cl.** ..... **126/110 R; 126/116 R; 165/DIG. 2; 237/55**

[58] **Field of Search** ..... 126/110 R, 110 B, 116 R, 126/99 R; 237/55; 165/DIG. 2; 431/1

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Primary Examiner—William F. O'Dea

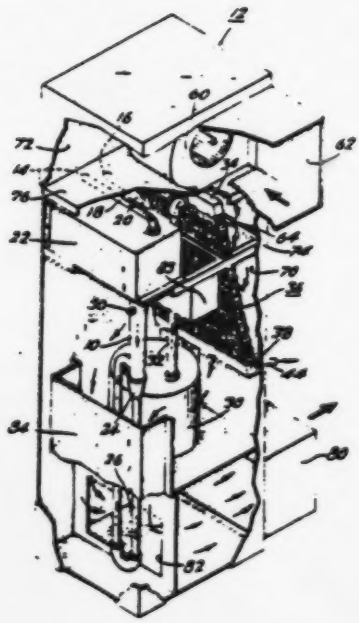
Assistant Examiner—William E. Tapolcai, Jr.

Attorney, Agent, or Firm—Howson and Howson

[57] **ABSTRACT**

A secondary heat exchanger is connected in the flow path of exhaust gases passing from the exhaust expansion chamber to the flue gas outlet of a pulse combustion system, so as to be heated by the hot exhaust gases flowing therein, and air to be heated is first passed over the exterior of the secondary heat exchanger and then over the exterior of the combustion chamber. The temperature at the outlet end of the secondary heat exchanger is below the condensation point of exhaust gases therein, and the temperature at the inlet end of the secondary heat exchanger is sufficiently low to permit use of a wider range of materials for the secondary heat exchanger. The secondary heat exchanger is made of a material of high thermal conductivity which is corrosion resistant, and increases the thermal efficiency of the system greatly while exerting a muffling action to reduce the noise in the system.

7 Claims, 5 Drawing Figures



U.S. Patent Aug. 14, 1979 4,164,210

FIG. 1.

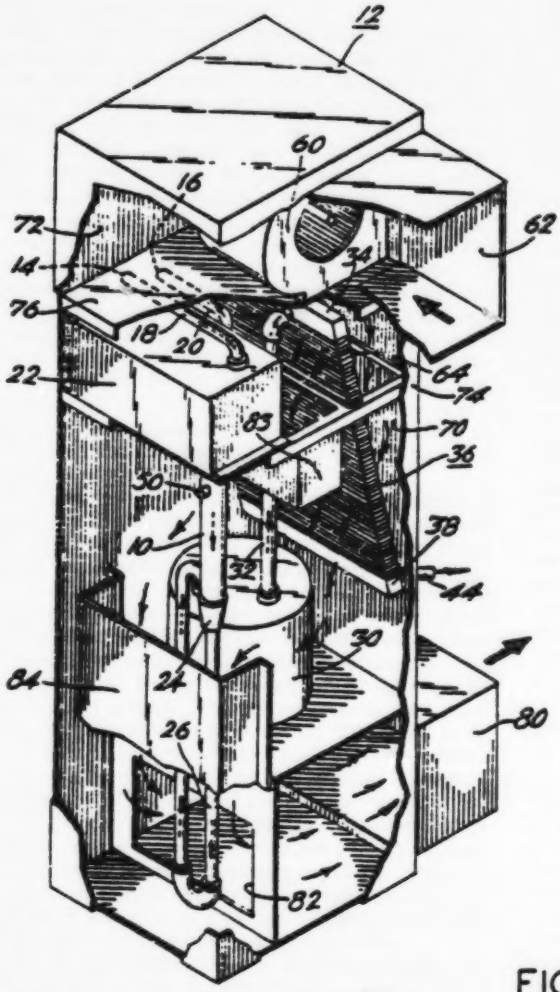


FIG. 2.

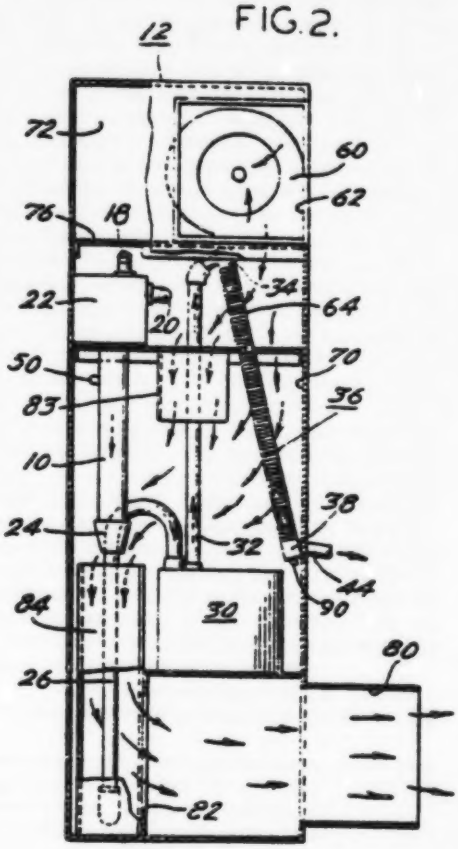


FIG. 3.

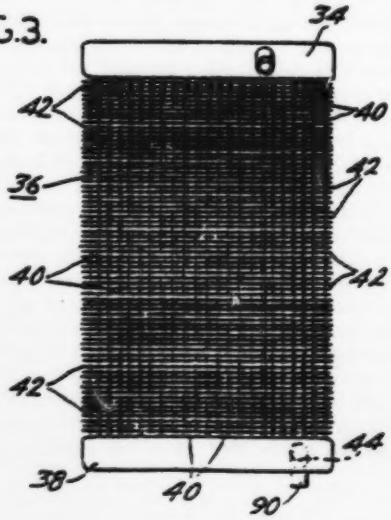


FIG. 4.

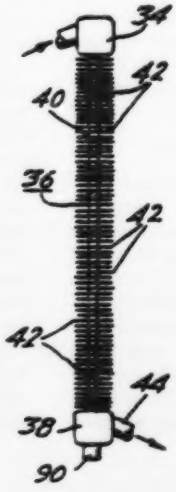
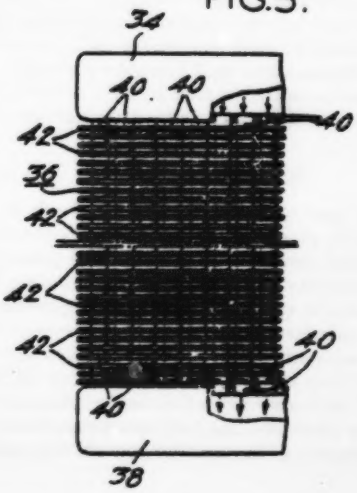


FIG. 5.



PULSE COMBUSTION SYSTEM FOR HEATING OF AIR

BACKGROUND OF THE INVENTION

This invention relates to improvements in pulse combustion systems for heating air.

Pulse combustion heater systems have been known for many years. In such devices, a combustible fuel and combustion air are admitted into a pulse-combustion chamber where they are ignited to produce an internal explosion, with resultant generation of heat. Immediately after each such explosion, an accoustically-produced negative pressure in the chamber draws additional air and fule into the chamber through appropriate valves, whereupon the next explosion occurs and closes the valves until the next negative pressure occurs. Accordingly, once started, a self-perpetuating series of heat-releasing explosions are produced, with combustion air and fuel being sucked in automatically and intermittently through appropriate air and gas inlet valves as needed. In response to the combustion chamber pulses of high pressure, the hot exhaust gases from the combustion chamber are normally expelled forcefully through a tail pipe leading to an exhaust decoupling or expansion chamber, from which an exhaust-pipe line extends to an exhaust flue outlet.

In a typical system then, air from outside a building is drawn into the combustion chamber along with combustible gas to produce the desired pulse combustion therein, the exhaust gases resulting from this combustion then passing through the tail pipe, decoupling chamber, an exhaust-pipe line and a flue outlet to the exterior of the building; this flow occurs without requiring special fans or blowers, in response to the pumping action provided by the acoustically-resonant combustion chamber and tail pipe, in combination with combustion-air and fuel-gas inlet valves. Room air to be heated may be passed by forced draft over the exterior of the combustion chamber, the exhaust expansion chamber and at least part of the tail pipe, and then returned in heated condition to the room. The combustion chamber may in some cases be provided with external fins to enhance the heat exchanger operation. Normally it is also desirable, or necessary, to provide one or more mufflers in the exhaust pipe line to reduce the noise generated by the pulse combustion operation.

U.S. Pat. No. 2,916,032 of J. A. Kitchen, filed Oct. 10, 1957 and issued Dec. 8, 1959, shows one previously-known pulse combustion system for accomplishing the heating of room air. In the system of this patent, the pulse combustion chamber a is supplied with combustion air and fuel gas, the combustion chamber is provided with external heat transfer fins b, the hot exhaust gases from chamber a pass through an exhaust manifold c to a heat exchanger consisting of a plurality of tubes d in zig-zag form having heat transmitting metal plates e combined therewith, and the tubes and plates are arranged parallel with each other in a plane parallel with the direction of the flow of the air to be heated. The outlet ends of the heat exchanger tubes are connected to another gas manifold f, from which the exhaust gases are supplied to a compartment r serving as an exhaust gas cushion. The exhaust gases from compartment r are conveyed through a perforated pipe u to the atmosphere, this pipe serving as an exhaust gas muffler. The flow of the air to be heated is from beneath the exhaust manifold f, across the latter, then across the exterior surfaces of the heat exchanger, and finally across the exterior surfaces of the finned combustion chamber into the space to which the heated air is to be delivered.

While suitable for many purposes, systems such as that shown in the above-referenced patent generally require one or more special exhaust mufflers to minimize what would otherwise constitute very objectionably loud noise, caused by the explosions in the pulse combustion chamber. They are also subject to the problem that at least the inlet end of the heat exchanger operates at extremely high temperatures and must therefore consist of materials especially adapted for such high temperature operation, while its lower or outlet end operates at greatly reduced temperatures for which condensation of exhaust gases occurs, leading to the possibilities of metal corrosion near the outlet end of the heat exchanger. The heat exchanger in such a system is therefore such as to require, for best operation, use of materials which are not only satisfactory for extremely-high temperature operation, but are also of high thermal conductivity and resistant to corrosion by condensate formed in the lower temperature portions thereof. Ferrous metals have the necessary high-temperature stability, but have rather poor thermal conductivity and corrosion resistance; copper and aluminum, for example, on the other hand have excellent thermal conductivity and corrosion resistance but tend to soften at very high temperatures. Accordingly, in such a prior art system the simultaneous requirements of high thermal conductivity for best heat exchange, high-temperature stability in the face of the very high temperatures at the inlet of the secondary heat exchanger, and corrosion resistance near the outlet of the secondary heat exchanger produce problems with respect to the materials to be used in making the heat exchanger.

It is an object of the present invention to provide a new and useful pulse combustion system for the heating of air.

Another object is to provide such system which provides high thermal efficiency by providing a relatively large amount of heat-exchanger surface.

A further object is to provide such a system in which the need for special exhaust mufflers is greatly reduced or completely eliminated.

Still another object is to provide such a system in which the heat exchanger operates at relatively low maximum temperatures, whereby the materials used therein need only be compatible with requirements of corrosion resistance and high thermal conductivity, and are not also subject to the severe and often conflicting requirements of operation at very high temperatures.

It is also an object to provide such a system which is compact and relatively inexpensive to make and maintain.

SUMMARY OF THE INVENTION

According to the invention, a secondary heat exchanger is connected in series in the flow path of hot exhaust gases passing from the exhaust expansion chamber to the flue gas outlet, the secondary heat exchanger preferably comprising, for example, a plurality of finned heat-conductive pipes of corrosion-resistant materials of high thermal conductivity, through which the exhaust gases pass. The air to be heated is passed first over the surfaces of the secondary heat exchanger, then over the surfaces of the combustion chamber, and preferably also over the tail pipe and the exhaust decoupling chamber.

Since the secondary heat exchanger, and especially the inlet end thereof, is subjected to a flow of the relatively cool newly-admitted room air, it operates at relatively-low temperatures, typically of the order of less than 150° F. at its outlet, and less than 800° F. at its inlet, and therefore while it provides useful heating of the newly-admitted air it can be made of materials which are not required to withstand extremely high temperatures. It also serves as an effective noise muffler, generally making other special exhaust mufflers unnecessary, thereby avoiding the cost as well as the bulk typically required by special exhaust mufflers. The temperature in the secondary heat exchanger near its outlet is preferably low enough that condensation occurs therein, resulting in high thermal efficiency of heating of the room air.

BRIEF DESCRIPTION OF FIGURES

These and other objects and features of the invention will be more readily understood from a consideration of the following detailed description, taken in connection with the accompanying drawings, in which:

FIG. 1 is a perspective view, with portions broken away, of a pulse combustion system suitable for the heating of room air, embodying the present invention in one of its forms;

FIG. 2 is a side view of the embodiment of FIG. 1, with portions broken away;

FIG. 3 is a face view of the secondary heat exchanger employed in the systems of FIGS. 1 and 2;

FIG. 4 is an end view of the heat exchanger of 3; and

FIG. 5 is an enlarged fragmentary view of portions of the heat exchanger of FIG. 3, showing the construction thereof in more detail.

DESCRIPTION OF SPECIFIC EMBODIMENTS

Referring now to the specific embodiment of the invention shown in the drawings by way of example only, a pulse combustion chamber 10 in the form of a relatively large diameter tube is mounted within a furnace housing 12, and supplied with fuel gas from fuel gas inlet 4 and with combustion air from combustion air inlet 16. The latter two inlets are connected by respective supply tubes 18 and 20 to an enclosure 22, which contains conventional fuel gas and air expansion chambers, gas and air flapper valves, and an auxiliary blower for use in initial start up, by means of which combustion air and fuel gas are supplied to the interior of pulse combustion 10. The details of these supply lines, the expansion chambers, the flapper valves and the auxiliary blower are not shown, in the interest of clarity, since they may be constructed in a variety of manners well known in the art and are not in themselves necessarily different from previously-known arrangements utilized for this purpose.

The downstream or exhaust end of pulse combustion chamber 10 is connected through a pipe reducer 24 to a tail pipe 26, the downstream or exhaust end of which leads into the top of an exhaust expansion chamber 30, sometimes known as a decoupling chamber. An exhaust pipe 32 leads from chamber 30 to the upper header 34 of a secondary heat exchanger 36.

The secondary heat exchanger 36 comprises the top header 34, a lower header 38 and a plurality of parallel, vertical, spaced-apart thermally-conductive pipes 40 each connected between the headers, together with an array of horizontally-extending thermally-conductive fins such as 42 each in contact with each of the pipes 40. The outlet 44 from the lower header 38 constitutes the flue outlet for the system, which may be connected to the exterior of the building, the room air of which is to be heated by the furnace here being described.

The portion of the system thus far described operates in the normal manner, with the exception of the function of the secondary heat exchanger. Thus, pulse combustion may be initiated in pulse combustion chamber 10 by means of an igniter 50, such as a spark plug, with the initial flow of combustion air and fuel gas into the chamber 10 provided by the initially-operated blower located within chamber 22 and the fuel gas suppled in response to the normal gas-line pressure; thereafter the burner operation is self-sustaining. The hot exhaust gases from the combustion chamber pass downstream through the tail pipe 26, the exhaust expansion chamber 30, the exhaust 32, the secondary heat exchanger 36 and the flue outlet 44 to the exterior.

At the same time, a main room-air blower 60 pulls in room air by way of conduit 62 and delivers it downwardly onto the outer face 64 of heat exchanger 36, the room air preferably initially flowing against the inlet end of the outer face of the heat exchanger as shown by the heavy room-air arrows in the drawings. The heat exchanger is positioned with its bottom and opposite sides in substantially air sealed relation to the interior housing walls 70, 72, and 74; the upper portion of secondary heat exchanger 36 is in substantial air sealing relation with a horizontal platform 76 positioned above it, on which main blower 60 may be mounted. Accordingly, the room air to be heated is constrained to flow through the openings between the pipes and fins in the secondary heat exchanger. The pulse combustion chamber 10, the tail pipe 26, the exhaust expansion chamber 30 and the exhaust pipe 32 are all positioned in the path of the room air flow, between the point at which it passes through the secondary heat exchanger 36 and the point at which it is delivered to the room-air discharge conduit 80, by way of conduit opening 82 near the bottom of the furnace housing 12. A U-shaped baffle 83 serves to divert a substantial part of the internal room air flow over the expansion chamber 30. A shield 84 mounted to the exterior walls of the furnace housing 12 assists in shielding the outer housing 12 from heat from the tail pipe 26.

It is noted that in this system there is no exhaust muffler other than the secondary heat exchanger 36, which provides this function in itself. It is also noted that the room air to be heated is constrained to pass first through the secondary heat exchanger 36 before it passes over the much hotter surfaces of combustion chamber 10, tail pipe 26 and expansion chamber 30. The secondary heat exchanger is thereby kept relatively cool, and in fact under preferred operating conditions is sufficiently cool to cause condensation of water in the exhaust gases within it, with the resultant recovery and transfer to the room air of the latent heat of vaporization of such gases. A condensate drain 90 is provided at the bottom of the secondary heat exchanger to permit continuous draining of such condensed fluid.

Since the secondary heat exchanger does operate at relatively low temperatures, it need not be made of special materials such as stainless steel, which are especially adapted for high temperature operation but unfortunately are of lower thermal conductivity than desired. Instead, it need merely meet the requirement of being stable at moderate temperatures and adequately corrosion-resistant in the presence of the condensate, and can therefore be of a corrosion-resistant material of

high thermal conductivity, for example, the vertical tubes 40 and the headers 34 and 36 may be of copper, and the fins 42 may be of aluminum.

Without thereby in anyway limiting the scope of the invention, the following examples of specific parameters of one system, constructed in accordance with the invention and the foregoing drawings, may be as follows. The entire furnace unit may be approximately 18 ½ inches wide, 21 ½ inches deep and about 58 inches high. The combustion chamber, tail pipe and exhaust pipe may be made from standard iron pipe fittings, the combustion chamber typically being an 8 inch length of 4 inch pipe; the combined length of combustion chamber and reducer 24 may be 12 inches. The tail pipe and exhaust pipe may each be of 1 ½ inch pipe; the length of the tail pipe 26, from the reducer 24 to the inlet to the exhaust expansion chamber 30, may be about 96 inches. The exhaust expansion chamber may be a cylinder 12 inches in diameter and 12 inches long. The exhaust pipe 32 may be an 18 inch length of 1 ½ inch pipe. The secondary heat exchanger 36 may comprise 13 30-inch long vertical copper tubes and the flue outlet pipe 44 might be 1 inch ID copper tube.

In such a system, the temperature of the flue gas entering the secondary heat exchanger is typically about 600° to 650° F. Halfway down the secondary heat exchanger, the temperature is typically about 140° F. At the exit through the one-inch diameter flue outlet tube, the flue gas temperature is typically about 95°–100° F. With such an arrangement, overall thermal efficiencies of about 85–98% have been obtained with satisfactory combustion, the operating level of CO_2 being typically about 10.0–10.5%, with a less than 0.01% air-free CO, for an input of 100,000 B.T.U.H. of natural gas.

With such maximum temperatures of about 650° in the heat exchanger, copper materials and usual solders can be used in the secondary heat exchanger, providing convenience in construction, high thermal conductivity and good corrosion resistance to condensed gases. While the condensate is primarily water in the case of natural gas fuel, it also typically contains some sulfur compounds, traces of other materials, and a mild acidity, due to impurities in the fuel gas, and it is these substances other than water which can be corrosive to certain materials such as ferrous metals.

It is preferred that the maximum temperature of the exhaust gases in the copper secondary heat exchanger be limited to no higher than about 800° F.; in some cases it may be desirable to limit it to very low values, such as 500° F. The exact operating temperature can be adjusted in design as a function of the heat output of the burner and the volume rate of flow of room air over the secondary heat exchanger, for example. In this connection it is noted that conventional control means will normally be used to assure that the room air flow is always present at the secondary heat exchanger whenever the temperature of the exchanger would otherwise become too hot. For example, the room air blower may be controlled always to come on when the burner starts up, and to remain on after the burner stops for a time long enough to reduce the maxiumum temperature of the secondary heat exchanger to the desired level when the blower is turned off.

In a conventional arrangement in which the exhaust gases in the combustion chamber of a pulse combustion system have a maximum temperature of about 1,800° to 2,000° F., it will generally be necessary to use a high-temperature resistant material such as ferrous metal for the heat exchanger; to reduce corrosive effects, stainless steel may be used. However, stainless steel and other ferrous metals will not provide the high thermal conductivity of copper, and the thermal efficiency of the system will therefore be reduced by the limited options available with respect to the material which can be used.

The above-mentioned condensation of water in the exhaust gases has been found to make it possible to increase the thermal efficiency to above about 89 or 90%; for example, while without condensation thermal efficiencies of up to about 80–89% can be achieved, about 95–98% efficiency can be obtained when condensation is caused to occur in the secondary heat exchanger. In the type of system described, condensation will generally first occur at a point in the secondary heat exchanger where the gas temperature has dropped to about 125°–150° F., and the outlet end portion of the secondary heat exchanger is preferably operated to produce gas temperatures below about 150° F. for this reason.

While the invention has been shown and described with particular reference with specific embodiments thereof, it will be understood that it may be embodied in a variety of forms diverse from those specifically shown and described, without departing from the spirit and scope of the invention as defined by the appended claims.

What is claimed is:

1. In an air-heating pulse combustion furnace system comprising a pulse combustion chamber, means for feeding combustion air and combustible gas into said chamber to form a combustible mixture, an exhaust expansion chamber connected to receive hot exhaust gases from the outlet of said combustion chamber, a tail pipe connected to said combustion chamber and supplying hot exhaust gases from said combustion chamber to said expansion chamber, a flue gas outlet, an exhaust pipe line connecting said expansion chamber to said flue gas outlet gas, means for igniting said combustible mixture to produce pulse combustion in said chamber, and means for directing a flow of air to be heated over the exterior surfaces of said combustion chamber to heat said air:

the improvement comprising a secondary heat exchanger connected in said exhaust pipe line between said expansion chamber and said flue gas outlet so as to be heated by said exhaust gases flowing therein; and

means for directing the flow of said air to be heated over the exterior surfaces of said secondary heat exchanger prior to said flow thereof over the exterior surfaces of said combustion chamber.

2. The system of claim 1, wherein said secondary heat exchanger comprises an array of pipe lines having fins secured thereto, said exhaust gases flowing through said secondary heat exchanger pipe lines, said secondary heat-exchanger pipe lines being of a high thermal conductivity material.

3. The system of claim 2, wherein said tubes are of a material selected from the class consisting of copper, aluminum and highly thermally conductive alloys thereof.

4. The system of claim 1, wherein the temperature of said exhaust gases and the flow of said air to be heated is such that the temperature of said exhaust gases near the outlet end of said secondary heat exchanger is below the condensation point for at least some compo-

nents of said exhaust gases, whereby condensate is formed in said secondary heat exchanger.

5. The system of claim 1, wherein the temperature of said exhaust gases at the inlet end of said secondary exchanger is no greater than about 800° F., and at the outlet end thereof is less than about 150° F.

6. The system of claim 1, wherein said gas is natural gas.

7. The system of claim 1, wherein said means for directing the flow of said air to be heated over the exterior of said secondary heat exchanger comprises blower means for creating said flow of said air to be heated, said secondary heat exchanger being positioned along the path of said flow nearer said blower means than is said pulse combustion chamber.

* * * * *

United States Patent [19]

Schossow

[11] **4,171,089**

[45] **Oct. 16, 1979**

[54] **HEAT EXCHANGER**

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[73] Assignees: George W. Schossow, White Bear Lake; Merchant, Gould, Smith, Edell, Welter & Schmidt, P.A., Minneapolis, both of N. Mex., part interest to each

[21] Appl. No.: 798,567

[22] Filed: May 19, 1977

[51] Int. Cl.[2] F24B 7/00

[52] U.S. Cl. ..... 237/55; 126/85 B; 126/117; 165/DIG. 2; 126/112

[58] Field of Search ..... 165/DIG. 2, 140; 126/117, 112, 85 B, 108; 122/20 B; 237/55

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Attorney, Agent, or Firm—Merchant, Gould, Smith, Edell, Welter & Schmidt

[57] **ABSTRACT**

A self-cleaning heat exchanger in combination with a home furnace system which draws combustion air from outside the home is disclosed. Gases of combustion are drawn up through a plurality of vertical tubes and out a chimney by a fan mounted in the chimney. The lower portions of the tubes are cooled by inside air drawn over them by the return air blower of the furnace, the upper portions of the tubes are cooled by cold outside air drawn across them by the furnace burner unit. Moisture which condenses in the upper portions of the tubes is drawn down through them by gravity. The higher temperature in the lower part of the tubes causes this condensate to revaporize and rise to the upper portion of the tubes, where the process repeats itself. As more water vapor enters the tubes in the flue gas, distilled condensate eventually drops out of the tubes and runs into a drain at the lower end of the heat exchanger. The moisture from condensation of the combustion gases scrubs pollutants from the flue gases passing through the tubes, and is slightly acidic, providing a cleaning function for the interior of the heat exchanger tubes. The heat exchanger is also suitable for use in commercial furnace systems and combustion-type power plants which utilize outside air for combustion.

4 Claims, 3 Drawing Figures

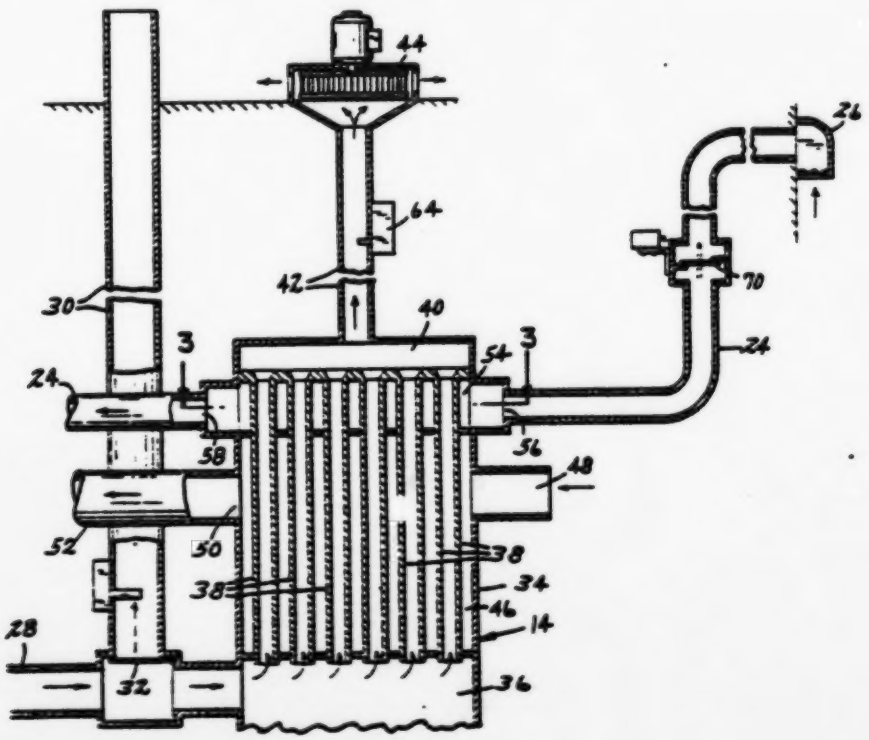


FIG. 1

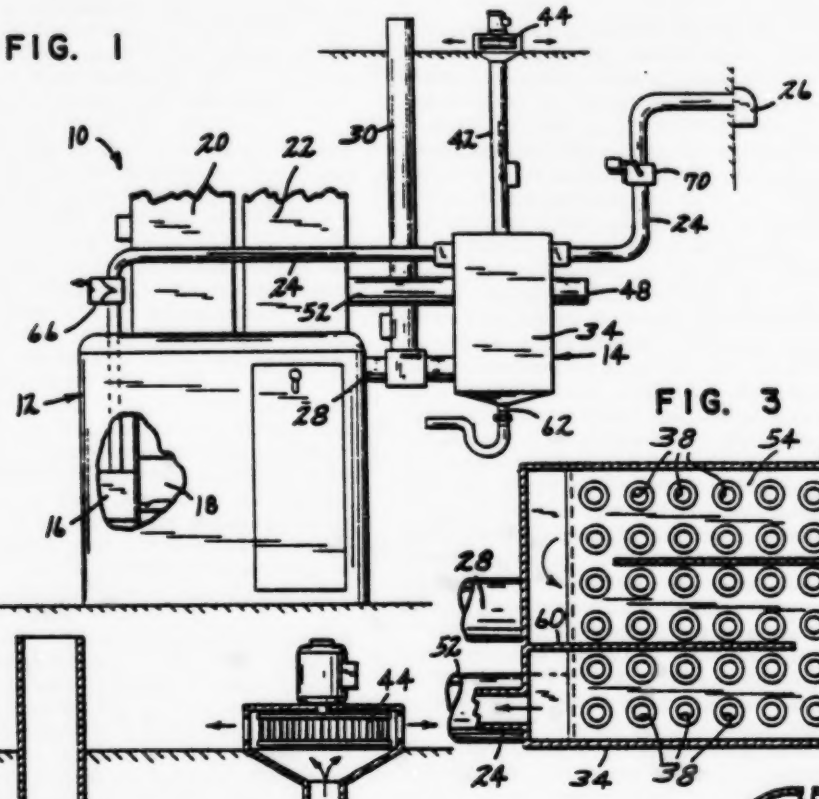


FIG. 3

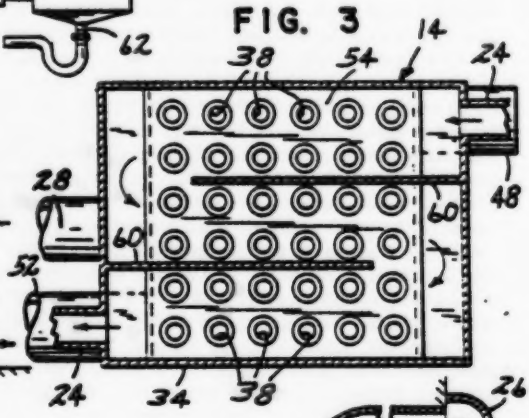
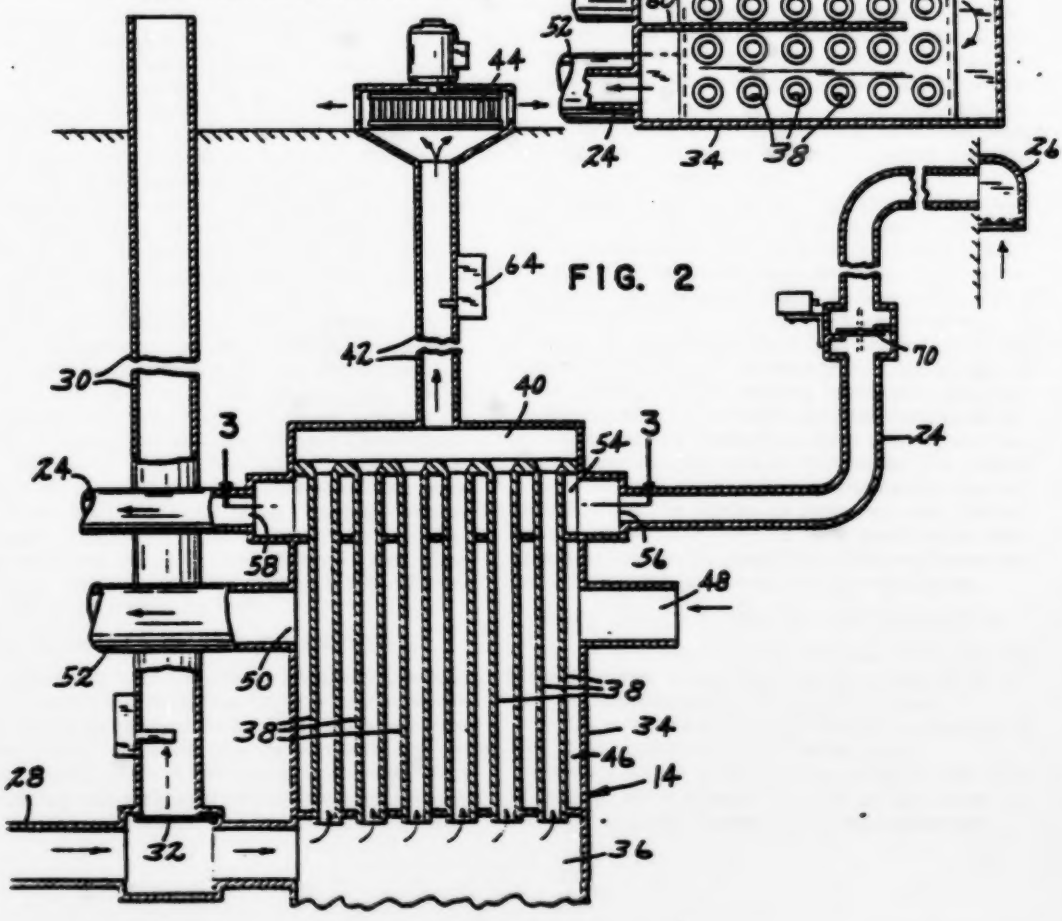


FIG. 2



HEAT EXCHANGER

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates generally to the field of heat exchangers used in combination with power plants, and commercial and residential heating systems. More particularly, this invention relates to the field of heat exchangers for use in home heating systems which draw cold air from outside the home for combustion.

2. Description of the Prior Art

As has been pointed out in my pending patent application Ser. No. 775,253, now abandoned home heating systems which utilize air from outside the home for combustion purposes offer significant advantages over conventional home heating systems, which draw combustion air from inside the home. Among these advantages are: (1) fuel savings due to the fact that heated room air is not being expelled up the chimney to be replaced by cold air seeping into the home; and (2) the elimination of the need for a humidifier to keep room air at a livable humidity level.

Because such systems draw cold air into their burner units rather than drawing air that has already been heated, the firebox temperature of such systems may be slightly lower than that of conventional systems. Additionally, as the stack gases proceed up the chimney, moisture contained in those gases condenses on the inside of the chimney, where it may cause chimney freeze up. Finally, the hot combustion gases being exhausted up the chimney represent wasted heat, which could be used for further room heating.

It is known in conventional furnace systems to utilize the hot stack gases for room heating by diverting the stack gases through a heat exchanger which distributes some of the heat from the flue gases to inside air. One such system is disclosed in U.S. Pat. No. 3,813,039. Heretofore, however, there has not been a furnace system which not only utilized the heat from flue exhaust gases to heat up air circulating within the home, but also was adapted to draw cold air from outside the home for combustion purposes, raise the temperature of that air slightly so as to increase the temperature of the firebox, and decrease the temperature and moisture content of the flue gases.

In the past, industrial combustion systems, such as coal-burning power plants, have utilized cyclone separators, electrostatic precipitators, and wet scrubbers to clean the stack gases of fly ash and pollutants. One of the problem pollutants has been sulpher, which is exhausted to atmosphere as sulpher dioxide. Power plants and commercial boilers thus utilize expensive, extremely tall chimney stacks to exhaust noxious gases well away from residential levels. Until the present invention, a system has not been known which satisfactorily scrubs the pollutants from the stack gases, eliminating the need for very high chimneys, and retrieves heat energy for subsequent work.

SUMMARY OF THE INVENTION

In accordance with the present invention, a furnace system suitable for residential use is provided, which draws air for combustion via a conduit extending from a point outside the home into the burner unit of the furnace. Along its route, the conduit is plumbed into a self-cleaning heat exchanger constructed in accordance with the present invention. The heat exchanger includes

a plurality of vertically oriented tubes which are in fluid communication at their lower ends with a conduit carrying the gaseous products of combustion away from the burner unit. At their upper ends, the tubes are in fluid communication with a chimney stack in the outlet of which a fan is mounted to induce a draft up the chimney.

The lower portions of the tubes extend through a first chamber which has an upstream vent to inside air and a downstream outlet which is plumbed to the return air plenum of the heating system. This arrangement allows room air from the vicinity of the furnace to be drawn across the tubes, thus picking up some heat before being passed into the circulation system of the furnace.

Near their upper ends, the tubes pass through a second chamber which has an upstream inlet attached to the outside air conduit, and a downstream outlet from which the combustion air conduit proceeds to the burner unit of the furnace. Thus, the cold air being drawn from outside the home by the burner unit passes through the upper chamber of the heat exchanger and circulates around the upper end portions of the gas-carrying tubes. Thus, the temperature of the cold combustion air is increased before combustion, so as to keep the temperature of the firebox at a preferred level without an excess use of fuel, and the hot flue gases are cooled to the point where the moisture carried by those gases condenses on the tube wall and runs down the tube walls. The stack gases proceed up the chimney and are exhausted from the home. As the condensate runs down the tubes, it revaporizes in the hot, lower portion thereof, and is carried back towards the top of the tubes, recycling and redistilling itself as the process continues. The continuous addition of more water vapor to the inside tube environment creates a relatively low dew point situation therein, and the upward passage of particulate matter through the tubes causes "rain" therein. This phenomenon in turn acts to scrub the exhaust gases of pollutants such as the SO_2, which reacts with the water to form sulphuric acid.

By virtue of the fact that the slightly acidic moisture from the stack gases condenses at the upper end of the heat exchanger tubes and runs down the tubes as a result of gravity, the heat exchanger tubes enjoy a self-cleaning feature. At the lower end of the heat exchanger, a drain collects the moisture run off from the tubes.

It will be appreciated from the foregoing that the present invention results in even greater fuel savings in home heating systems drawing combustion air from outside the home. The warming of the combustion air before combustion results in a higher temperature firebox without a greater rate of combustion. The utilization of the stack gases to heat up circulating room air further lessens the burden on the burner unit. The extraction of moisture from the stack gases before their exhaust eliminates the possibility of chimney freeze up, and acts to scrub pollutants from the stack gases.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a view in side elevation, with portions thereof broken away, showing the furnace/heat exchanger combination of the present invention.

FIG. 2 is an enlarged view in section of a portion of FIG. 1, with portions thereof broken away.

FIG. 3 is a view in section taken along the line 3—3 of FIG. 2, and indicating the flow of combustion air through the upper chamber of the heat exchanger.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring first to FIG. 1, there is disclosed a furnace/heat exchanger combination, generally designated by the numeral 10, in which a furnace 12 is coupled with a heat exchanger 14 constructed in accordance with the present invention. Furnace 12 comprises a burner unit 16, including a blower or fan (not shown) which draws air to be mixed with the fuel for combustion. A flame burns in firebox 18 to heat air which is subsequently distributed through the home via duct 20, and which is returned to the furnace for reheating via duct or plenum 22.

Burner unit 16 draws combustion air from outside the home via conduit 24, which extends from burner unit 16 to vent 26, which is located outside the home, the gaseous products of combustion are exhausted from furnace 12 via exhaust conduit 28, and may be selectively exhausted up chimney stack 30 or routed through heat exchanger 14. A manually operated bypass gate 32 is used to select which route exhaust gases will take.

Exhaust gases enter heat exchanger 14 when gate 32 closes chimney 30, as clearly shown in FIG. 2. Heat exchanger 14 comprises an outer wall 34 which encloses four chambers. The lowermost chamber 36 is a plenum into which the gaseous products of combustion flow via exhaust conduit 28. A plurality of vertically oriented tubes 38 are in fluid communication with lower chamber 36. Heat exchanger tubes 38 are of a type known in the art, such as those constructed with a copper outer wall and a corrosion-resistant inner wall of ceramic or teflon. As indicated in FIG. 2, the exhaust gases flow upwardly through tubes 38. At their uppermost ends, tubes 38 are in fluid communication with chimney stack 42. Mounted in the mouth of chimney stack 42 is a driven fan 44. The action of heat exchanger 14 causes the flue gases to cool markedly before reaching chimney 42, thus reducing the volume of the flue gases by 50% to 70%, with the result that fan 44 need not be a large power consuming fan in order to create the desired negative pressure in chimney 42. Since fan 44 is mounted downstream of heat exchanger 14, a negative pressure is created within tubes 38, which induces a draft upwardly through the heat exchanger so as to draw the exhaust gases through exhaust conduit 28, into lower chamber 36, through tubes 38 into chamber 40, and upwardly through chimney 42, and thus out of the house.

There are two intermediate chambers in heat exchanger 14. The lowermost of these chambers 46 has an upstream inlet 48, which is open to room air. Chamber 46 has a downstream outlet 50 which is in communication with the return air plenum 22 of the furnace via duct 52. The return air blower of the furnace (not shown) thus draws air from inside the home in the vicinity of the heat exchanger through chamber 46, where it circulates around tubes 38 before passing into the recirculation system of the furnace.

The uppermost intermediate chamber, indicated by the numeral 54, has upstream inlet 56 and downstream outlet 58 into which combustion air conduit 24 enters and exits.

Both chambers 46 and 54 are partitioned by vertically extending baffles 60, as shown in FIG. 3, so as to insure complete circulation around tubes 38 of the air entering and exiting those chambers.

Heat exchanger 14 includes a drain 63 located at its lower end to drain away the moisture which condenses and runs down tubes 38 into chamber 36.

Mounted in chimney stack 42 is a pressure sensitive switch 64 which is operably connected to burner unit 16 by means not shown, but well known in the art, and is adapted to prevent burner unit 16 from igniting, and to turn burner 16 off, if the appropriate negative pressure does not exist within chimney stack 42, i.e. if a draft up through the heat exchanger does not exist. Combustion air conduit 24 also includes the safety devices discussed in my pending patent application previously referred to, including a barometric draft regulator 66 mounted in a tee section of conduit 24 shown in FIG. 1, and a butterfly valve 70 which is normally open, but closes after a time delay following shut off of burner unit 16.

In operation, thermostatic control means (not shown) turn on fan 44 and open butterfly valve 70. When the appropriate draft up chimney 42 is sensed by switch 64, burner unit 16 is ignited. Cold air from outside the home is drawn into burner unit 16 via conduit 24 for combustion purposes. This cold air circulates through chamber 54 of heat exchanger 14 before passing into burner unit 16. The gaseous products of combustion are exhausted from firebox 18 via conduit 28 into lower chamber 36 of heat exchanger 14, where they are drawn up through tubes 38 and exit via chimney stack 42. The return air blower (not shown) of furnace 12 draws air from inside the home in through chamber 46 of heat exchanger 14 before entering the recirculation system of furnace 12. As the hot stack gases pass upwardly through tubes 38, cool inside air circulates around their lower portions where it picks up heat from them. At this point, the slightly acidic moisture carried within the exhaust gases condenses on the walls of tubes 38 and is drawn by gravity downwardly into chamber 36 and carried away via drain 62. This action of the moisture keeps tubes 38 clean. The cold combustion air travelling through conduit 24 is heated as it passes through chamber 54 and thus passes into burner unit 16 in a warmed state.

It will be appreciated from the foregoing that I have prevented a system which utilizes outside air for combustion purposes, thus having the fuel saving and humidifying advantages previously discussed, which utilizes hot stack gases to warm the combustion air slightly so as not to require a greater rate of combustion to keep the firebox at its desired temperature, and which utilizes the hot stack gases to preheat some of the recirculating air home. In addition, the cool air circulating through the heat exchanger cools down the stack gases and removes the moisture from them. Finally, the heat exchanger of the invention scrubs the flue gases of pollutants, particularly SO_2, and of particular matter, and is self-cleaning, requiring little or no maintenance.

What is claimed is:

1. The combination of a home furnace system wherein the furnace burner draws combustion air from outside the home with a heat exchanger which cools stack gases and removes moisture therefrom before the stack gases are exhausted from the home, warms outside combustion air before it reaches the burner, and warms cool inside air before recirculating it through the furnace, said heat exchanger comprising:

(a) a plurality of vertically oriented tubes having lower ends in fluid communication with a conduit carrying stack gases from the furnace, and upper ends in fluid communication with a chimney;

(b) a combustion air chamber which extends around portions of said tubes adjacent their upper ends, said combustion air chamber being in fluid communication with a source of cold air exterior to the home and with the furnace burner;

(c) a cold return air chamber which surrounds portions of said tubes adjacent their lower ends, said return air chamber having a downstream outlet in fluid communication with the return air blower, and an upstream inlet in fluid communication with the ambient inside air of the home;

(d) a fan mounted in the chimney for inducing a draft up the chimney; and

(e) means for draining moisture of condensation which runs down the tubes.

2. The furnace-heat exchanger combination of claim 1 which further includes pressure sensitive control means mounted in the chimney and operably connected to the furnace burner for igniting the burner when the fan creates a negative pressure within the chimney, and for turning the burner off when no such negative pressure is present.

3. The furnace-heat exchanger combination of claim 2 which further includes means for bypassing the heat exchanger and diverting the stack gases to a second chimney.

4. A heat exchanger suitable for use in home furnace systems wherein the furnace burner draws combustion air from outside the home, and in commercial systems wherein outside air is used for combustion by a burner, comprising:

(a) a combustion air chamber having an upstream end in fluid communication with a first conduit carrying outside air and a downstream end in fluid communication with a second conduit carrying air from the chamber to the furnace burner;

(b) a plurality of vertically oriented tubes, portions of which extend transversely through the combustion air chamber, said tubes having bottom ends in fluid communication with a conduit carrying exhaust gases from the burner, and having top ends in fluid communication with a chimney;

(c) means for inducing a draft up the chimney; wherein

(d) the portions of said tubes which extend through the combustion air chamber are adjacent the top ends of said tubes; and

(e) a return air chamber in fluid communication with a return air conduit of the system, said return air chamber being positioned below the combustion air chamber; wherein

(f) the bottom ends of said tubes are adjacent means for draining condensation liquids from the heat exchanger.

* * * * *

United States Patent [19]

Gary

[11] 4,176,787

[45] Dec. 4, 1979

[54] **HEAT RECOVERY DEVICE FOR USE IN RETURN AIR DUCT OF FORCED AIR FURNACE**

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[21] Appl. No.: 891,171

[22] Filed: Mar. 29, 1978

[51] Int. Cl.[2] F24B 7/00

[52] U.S. Cl. 237/55; 138/38; 165/179; 165/DIG. 2

[58] Field of Search ..... 237/55; 165/DIG. 2, 165/179; 122/20 B; 138/38

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Assistant Examiner—William E. Tapolcai, Jr

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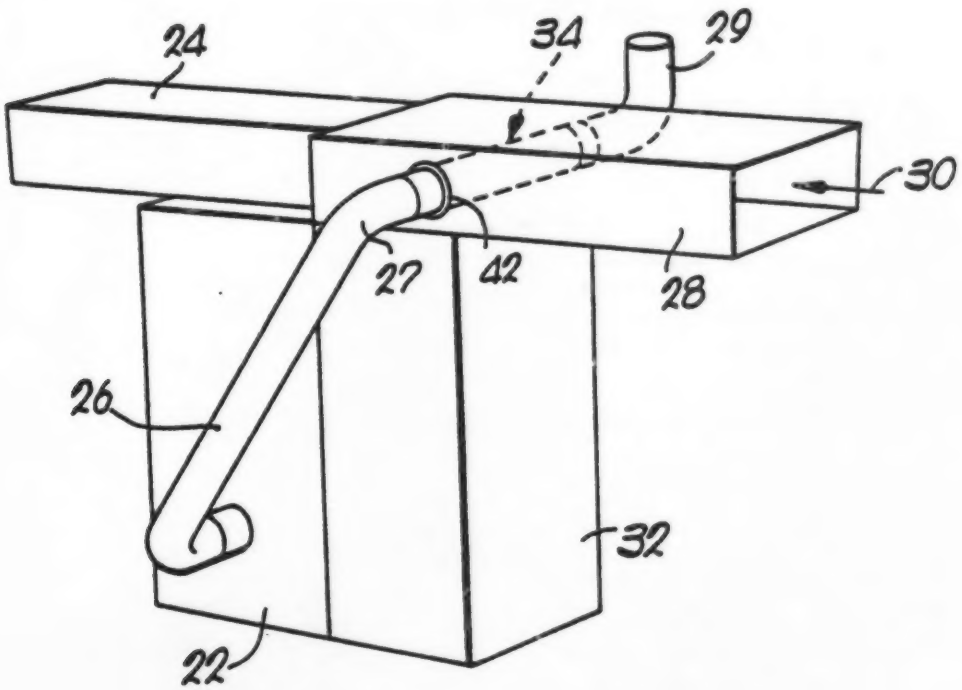
[57] ABSTRACT

An especially compact, highly efficient heat recovery device for use in the cool air return duct of a heating system may be easily installed in existing flue pipes and transfers the ordinarily wasted heat in the exhaust gases flowing through the flue pipe to the cool return air thereby preheating the latter and increasing the efficiency of the heating system. The device includes a three stage heat transfer structure housed within a heat conductive tubular member which cooperate to transfer heat away from the gases into the cool return air by means of conduction, convection and radiation processes. A first stage deflects a portion of the gases toward and into heat exchanging contact with the sidewalls of the tubular member and directs the remaining portion into a gas pervious heat storage trap which comprises the second stage. The third stage includes a perforated, heat deflecting and radiating cone structure which cooperates with the first and second stages to produce temperature stratification within the tubular member to further increase heat transfer to the cool return air.

12 Claims, 10 Drawing Figures



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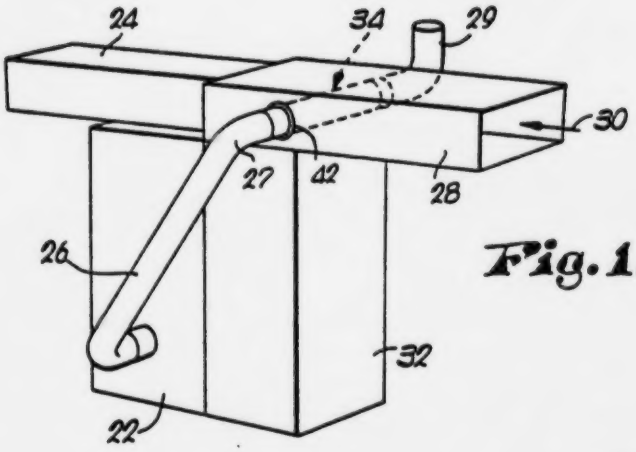
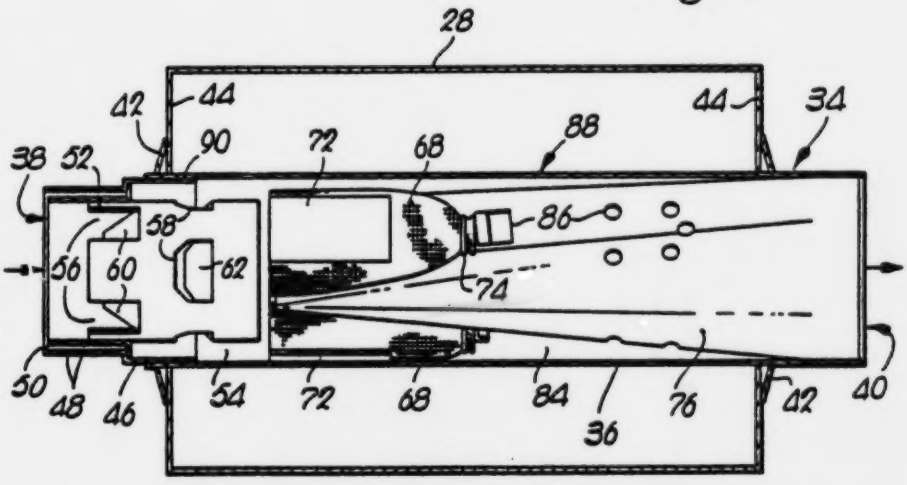


Fig. 1.

Fig. 2.



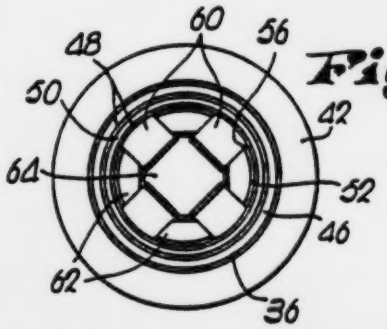


Fig. 3.

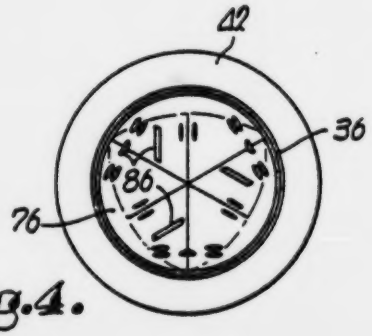


Fig. 4.

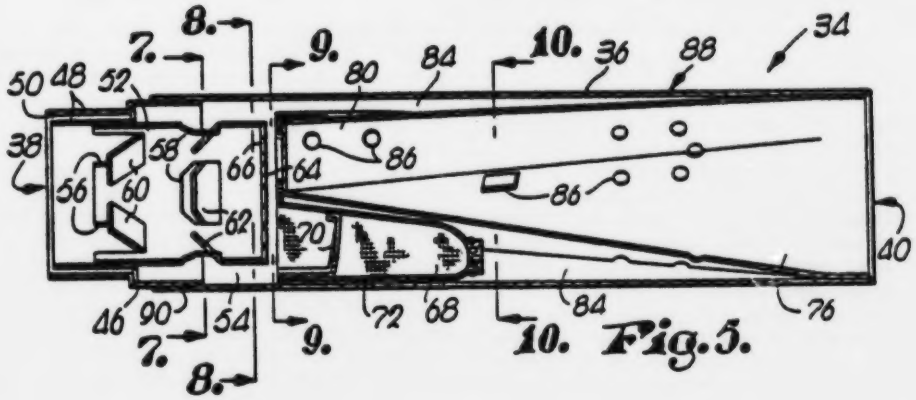


Fig. 5.

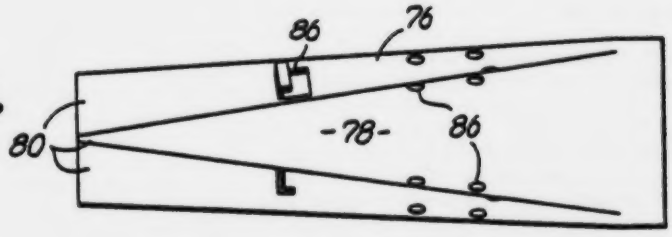


Fig. 6.

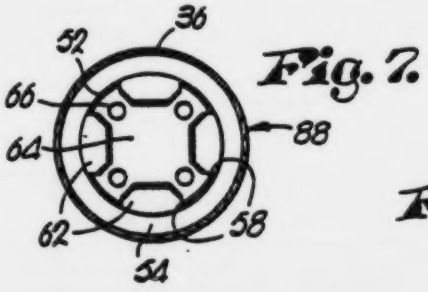


Fig. 7.

Fig. 9.

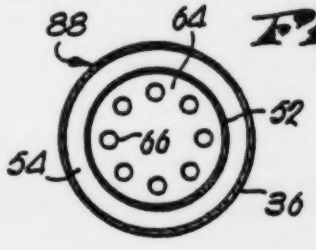


Fig. 8.

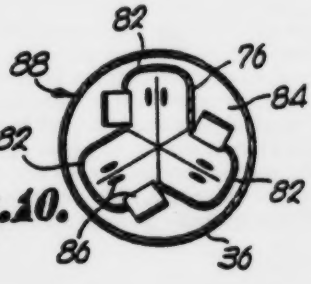


Fig. 10.

HEAT RECOVERY DEVICE FOR USE IN RETURN AIR DUCT OF FORCED AIR FURNACE

TECHNICAL FIELD

This invention generally relates to apparatus for exchanging or transferring heat, and deals more particularly with a device for use in the return air duct of forced air type furnaces for recovering and effectively utilizing heat from the flue of such furnace.

BACKGROUND ART

The growing concern over energy conservation prompted by limited natural resources has produced a need in the art for further improvements in the design of heating systems which increase operating efficiency and thereby reduce energy consumption. In combustive type heating systems, particularly those burning fossil-type fuels such as coal, gas, or oil, products of combustion and toxic gases are generated and must be drawn away and exhausted from the heating system into the open atmosphere, usually by means of a conduit or "flue pipe". As an energy wasting consequence of this type of system, the exhaust gases include a significant quantity of heat produced by the combustion process which is ordinarily lost to the atmosphere and therefore is not effectively employed for, and does not contribute to, the heating of the area which the heating system is intended to warm.

Others in the past have recognized the important advantage of recovering heat from the exhausted combustion gases, and have attempted to devise means for transferring a small portion of such exhausted heat to the area intended to be heated. These prior art systems are typically employed in forced air-type heating systems for buildings, and include a heating chamber provided with a warm air delivery duct and a cool air intake or return duct while a flue pipe for carrying away the gases and products of combustion is usually in communication with the heating chamber. The flue pipe, which normally comprises a metallic, heat conducting material, is disposed within the cool air return duct so that the cool air returning to the heating chamber passes over the warm surface areas of a short stretch of the flue pipe and is, in effect, slightly preheated prior to return to the heating chamber. In this manner, the air within the heating chamber to be heated is slightly increased, consequently, the energy required to elevate the temperature of the preheated return air to the desired temperature is slightly reduced.

Prior art systems of the general type described above are shown in U.S. Pat. Nos. 2,225,181 and 2,439,109. U.S. Pat. No. 2,225,181 discloses a rather large, concentric shell structure interposed between stretches of the flue pipe and in the path of cold air returning to the furnace through a return air duct, which is designed to transfer a portion of the heat flowing through the flue pipe solely by means of heat conduction and relies on a substantial surface area to maximize the heat transfer process; the apparatus shown by this patent is of limited effectiveness in recovering the wasted heat energy, particularly in view of the substantial size and space requirements for the heat recovering structure. U.S. Pat. No. 2,439,109 also shows the use of heat recovery structure interposed between stretches of a flue pipe disposed within the path of cool air flow in the return duct of a furnace, and includes radiating members presenting a relatively large surface area to the cool return air flowing thereacross in order to maximize the transfer of heat from the gases within the flue by conduction processes to the cooler return air; the heat recovery structure disclosed by this patent requires the use of a baffle means for diverting the combustion gases in the main stream of the flue pipe away from the latter and into the radiating structures. The apparatus disclosed by the latter mentioned patent is also undesirable from the standpoint that the heat exchanging structure is rather large in comparison to the quantity of heat which it recovers, and necessitates the provision of a cool air return duct having a relatively large cross-sectional area to accommodate the same.

DISCLOSURE OF THE INVENTION

A small, compact, but highly efficient heat recovery device adapted for simple and rapid installation between stretches of a flue pipe includes a straight tubular member extending through spaces within the cool air return duct of a furnace for transferring a substantial portion of the heat in the combustion gases to the cool return air flowing over exterior surface portions of the tubular member. The tubular member houses heat transferring structure including a first, second, and third stage which sequentially operate on the combustion gases flowing therethrough, by means of conduction, convection, and radiation, in order to extract a maximum quantity of the heat in such gases and transfer such heat to the exterior surface areas of the tubular member, and are novelly arranged within the tubular member to produce stratification of the heat in the gases flowing through the member.

The first stage of the heat recovery device comprises deflection structure for diverting a portion of the gas flow toward and into heat conducting contact with the circumferential interior walls of the tubular section, and further includes a cylindrical sleeve coaxially located within the tubular member and slightly spaced from the sidewalls of the latter which confines the flow of the diverted portion of the gas and creates an entrance friction which further contributes to heat transfer between the gases and the cool return air. A perforated partition enclosing one end of the cylindrical sleeve of the first stage reduces the flow volume of the gases to the second stage and produces a pressure drop in going from the first stage to the second stage while directing a portion of the flow of the gases into the second stage. The second stage is disposed downstream from the first stage and comprises a plurality of longitudinally extending, gas pervious, mesh-like structures circumferentially spaced around the interior walls of the tubular member and in heat conductive relationship to the latter. The third stage of the device comprises a longitudinally extending, generally hollow, conically shaped baffle structure having the base end thereof disposed downstream from the first and second stages and contacting the interior sidewalls of the tubular member in order to further reduce the rate at which the gases are allowed to flow through the tubular member. The mesh-like structures of the second stage are nested between the tapering sidewalls of the third stage baffle structure and the interior sidewalls of the tubular member. The tapering sidewalls of the baffle structure function to reflect and radiate heat stored within the mesh-like structures in a direction toward the interior surface portions of the tubular member, while a plurality of perforations are provided in such tapering sidewalls to allow the gases to

be drawn into the baffle structure, and out the open base thereof away from the tubular member.

By this compact three stage arrangement of heat transferring units, an increased quantity of heat may be transferred from the flue gases to the cool return air in comparison to known prior art devices, consequently, the present arrangement may be disposed within the confines of existing flue pipes presenting a relatively small exterior surface area of the flow of cool return air and obviates the need for bulky heat exchanging structures presenting large surface areas to the flow of cool return air heretofore required.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is a perspective view of portions of a conventional air heating furnace showing the heat recovery device which forms the currently preferred embodiments of my invention installed between stretches of a flue pipe and disposed within the cool air return air duct of the furnace;

FIG. 2 is a longitudinal, sectional view of the preferred form of the invention shown in FIG. 1, depicting the device in its mounted position within the return air duct, arrows indicating the direction of gas flow through the device;

FIG. 3 is a view of one end of the device shown in FIG. 2 through which gases enter;

FIG. 4 is a view of the opposite end of the device shown in FIG. 2 from which gases exit;

FIG. 5 is a longitudinal, cross-sectioned view of the device shown in FIG. 2;

FIG. 6 is a longitudinal view of the third stage unit of the device;

FIG. 7 is a cross-sectional view taken along the line 7—7 in FIG. 5;

FIG. 8 is a cross-sectional view taken along the line 8—8 in FIG. 5;

FIG. 9 is a cross-sectional view taken along the line 9—9 in FIG. 5; and

FIG. 10 is a cross-sectional view taken along the line 10—10 in FIG. 5.

BEST MODE FOR CARRYING OUT THE INVENTION

Turning attention first to FIG. 1, a furnace system generally indicated by the numeral 20 includes a heating chamber 22, wherein means (not shown) are provided for burning a fuel such as gas, oil, coal, or the like. A warm air delivery duct 24 is in communication with the heating chamber 22 and functions to carry the air warmed within the latter to an area to be heated, while a flue pipe 26 having one end thereof also in communication with the heating chamber 22 is provided to separate the gases and other products of combustion produced by the fuel burning process from the warm air delivered in the duct 24, and functions to carry the same away from the heating chamber 22 for exhaust. A cool air intake duct 28, typically comprising a return air duct in communication with the area to be heated, receives intake air in the direction of the arrow 30 and delivers the cool air to an intake plenum which is in airflow communication with the heating chamber 22. Air moving means (not shown) may be provided within the air plenum 32 for drawing cool air through the cool air duct 28 into the heating chamber 22 and forcing the warmed air out through the warm air duct 24. The flue pipe 26 extends through opposite sidewalls of the cool air duct 28, and has a heat recovery device, generally indicated by the numeral 34, interposed between stretches 27 and 29 of the pipe 26 and within the cool air duct 28. Thus, it can be appreciated that cool air flowing in the direction 30 through the cool air duct 28 flows across exterior surface areas of the heat recovery device and is then drawn through the air plenum 32 into the heating chamber 22, thence into the warm air duct 24 for delivery to the area to be heated.

Referring now also to FIGS. 2 through 10, the heat recovery device 34 comprises an elongate, tubular member 36 of generally circular cross section comprising a heat conductive material such as ordinary sheet metal and includes a gas inlet opening 38 and outlet opening 40 on opposite extremities thereof which openings are respectively coupled with stretches 27 and 29 of the flue pipe 26. In the preferred form of the device 34, the interior surface portions of the tubular member 36 are coated with a heat resistant, black finish to enhance the absorption of heat. The tubular member 36 includes a pair of longitudinally spaced, annular flanges 42 therearound and secured thereto which engage exterior surface portions of the opposed sidewalls 44 of the cool air duct 28, thereby mounting the heat recovery device 34 within the cool air duct 28. An annular reducing flange 46 is secured to the tubular member 36 inside the latter adjacent the inlet opening 38, and includes an attenuated neck portion consisting of a pair of annular, concentric sleeves 48 of differing diameters defining an annular, recessed area 50 for receiving the end of the stretch 27 of the flue pipe 26 therewithin. A generally cylindrically shaped, sleeve or shell member 52, made from a heat conducting material such as ordinary sheet metal, is disposed within the tubular member 36 adjacent the inlet opening 38 and is suitably secured to interior surface portions of one of the sleeves 48. Shell member 52 has the longitudinal axis thereof essentially aligned with the like axis of the tubular member 36, the end of the shell member 52 adjacent the inlet opening 38 being essentially open to receive gases therethrough. Exterior surface portions of the shell member 52 are spaced from interior surface portions of the tubular member 36 and form an annular passageway 54 for confining the flow of gases between the exterior of the shell member 52 and the interior sidewalls of the tubular member 36.

The shell member 52 includes a first set of circumferentially spaced openings 56 in the sidewalls thereof adjacent the inlet opening 38, while a second set of circumferentially spaced openings 58 are also provided in the sidewalls of shell member 52, in central longitudinal stretches of the latter, which place the passageway 54 in gas flow communication with the inlet opening 38. A first set of deflector elements 60 in the nature of rectangularly shaped blades which may be formed integral with the shell member 52 are in respective communication with corresponding ones of the openings 56 and extend axially inward toward the longitudinal axis of the shell member 52, while a second set of similarly configured deflection elements 62 are in communication with the respectively corresponding openings 58, and likewise extend inwardly toward the interior space within the shell member 52. As best seen in FIGS. 2 and 3, the first and second sets of openings 56 and 58 respectively, along with their corresponding sets of deflector elements 60 and 62 are circumferentially staggered or offset with respect to each other, whereby the deflector elements 60 and 62 present deflecting structure to the

flow of gas through the tubular member 36 which essentially circumscribes the entire interior surface areas of the shell member 52, but which allows an axially central portion of gas flow to pass through the shell member 52 essentially undisturbed. A circular plate 64 is suitably secured to, and essentially encloses, the end of the shell member 52 distal from the inlet opening 38. Circular plate 64 includes a plurality of circumferentially, and essentially equally radially spaced apertures 66 therethrough placing interior spaces within the shell member 52 in gas flow communication with the outlet opening 40. Shell member 52, including openings 56 and 58, as well as deflector elements 60 and 62 and circular plate 64 form, in combination with the tubular member 36, a first stage in the heat recovery device 34 for effecting the transfer of heat from the gases to the cool air flowing through the cool air duct 28.

A heat storing trap, comprising the second stage of the heat recovery device 34 for effecting the transfer of heat, includes a plurality of circumferentially spaced, longitudinally extending, tubular heat trap structures 68 each having one end thereof distal from the inlet opening 38 essentially closed while the opposite extremity of each thereof proximal to the circular plate 64 is essentially open. Apertures 66 in the circular plate 64 are radially aligned with the open ends of the trap structures 68. The trap structures 68 each comprise an essentially elliptically shaped cross section, as best seen in FIG. 9, and are made from a heat conducting, gas pervious material such as a wire mesh which freely allows gases to pass therethrough. A gas deflection plate 70 having a V-shaped cutout portion therein is disposed within central longitudinal stretches of each of the tubular trap structures 68. Each of the trap structures 68 includes exterior surface portions aligned with, and closely adjoining, the interior surface portions of the tubular member 36 and are secured to the latter by means of a longitudinally extending, heat conducting plate member 72 which conformally engages the interior surface portions of the tubular member 36 and is joined in heat conducting relationship to the latter as well as to the trap structure 68. A fastening means 74 may be employed to crimp one end of the trap structure 68 whereby to close such end.

The third stage of the heat recovery device 34 comprises a baffle structure 76 made as by forming from a heat conductive material such as sheet metal or the like. Baffle structure 76 includes longitudinally extending, tapering, inclined sidewall portions 78 which form an essentially hollow, generally conically shaped member, having the smaller end thereof distal from the outlet opening 40 essentially closed while the larger base end thereof adjacent the outlet opening 40 is essentially open. The baffle structure 76 includes a plurality of longitudinally extending fins 80 equally circumferentially spaced around the exterior surface areas of the inclined sidewall portions 78, which extend radially outwardly from the longitudinal axis of the baffle structure 76 and function to partially partition the space within the tubular member 36 surrounding the inclined sidewall portions 78 into three spaces of essentially equal volumes. As best seen in FIG. 10, the fins 80 have the longitudinal axes thereof aligned with, and defined by, each of three respectively associated nodes 82 formed by the tapering of the inclined sidewall portions 78. As may be viewed in FIG. 4, the base end of the baffle structure 76 comprises an essentially circular cross section having an outside diameter marginally smaller than the inside diameter of the tubular member 36 and is joined to the latter adjacent the outlet opening 40 thereby preventing the escape of gas through the outlet 40 between the spaces 84 defined between the interior surface portions of the tubular member 36 and the baffle structure 76. The baffle structure 76 includes a plurality of perforations 86 in the sidewall portions 78 and fins 80 thereof, placing the spaces 84 surrounding the exterior of the baffle structure 76 in gas flow communication with the essentially hollow interior space thereof, and thus in communication with the outlet opening 40. As best seen in FIGS. 5 and 9, the elongate trap structures 68 are interposed between the inclined sidewall portions 78 and interior surface portions of the tubular member 36, and are nested in the spaces 84 defined by the fins 80. Portions of the exterior sidewalls of the trap structures 68 conductively contact the inclined sidewall portions 78 and fins 80 of the baffle structure 76.

The heat recovery device 34 may be simply installed by first positioning the tubular member 36 through openings of suitable diameter in opposing sidewalls of the cool air duct 28 and fastening the annular flanges 42 against such sidewalls to securely hold the device 34 in place. Stretch 27 of the flue pipe 26 may then be inserted into the recessed area 50 adjacent the inlet opening 38 of the device 34 and suitably secured in place therewithin. Similarly, stretch 29 of the flue pipe 26 may be slipped over the end of the tubular member 36 adjacent the outlet opening 40 and held there in place by any suitable means.

In operation, the heat recovering device 34 provides a three stage structure for maximizing the transfer of heat from gases flowing through the flue pipe 26 to the cooler air within cool air duct 28 flowing across the exterior surface portions of the tubular member 36 disposed within the duct 28. It can be readily appreciated, however, that the heat recovery device 34 is especially compact and houses the highly efficient three stage heat transferring structure within a tubular member 36 which is of essentially the same diameter of the flue pipe 26, so that the device 34 may be installed in existing heating systems without the need for substantial alteration of heating or cooling ducts to provide additional space for accommodating the same. Assuming now that cool air is being drawn in the direction 30 across the exterior surface areas of the tubular member 36, and into the heating chamber 22 for heating, gases entering the device 34 through the inlet opening 38 first flow into contact with the first set of deflector elements 60 resulting in a portion of the gas flow being diverted through the openings 56 into the annular passageway 54, while a further portion of the gas flow is similarly diverted by the second set of deflector elements 62 through the second set of openings 58 into the passageway 54. As the diverted portion of the gas flow enters the passageway 54, a slight compression of the gases therewithin results, thereby increasing the conductive heat transfer between the gases within the passageway 54 and the interior surface portions of the tubular member 36 adjacent the shell member 52. Moreover, intimate contacting of the flowing gases with interior and exterior surface portions of the shell member 52 result in the conductive transfer of heat from the gas flow to the shell member 52, which heat is inturn conductively transferred by the member 52 to the interior surface portions of the tubular member 36 at the point of surface contact indicated by the numeral 90. The circular plate

64 functions to restrict the flow of undeflected gases. The apertures 66 in the plate 64 function to direct a major portion of the undeflected gas flow into the open ends of the heat trap structures 64, a minor portion of such undeflected gas flow being directed into the spaces 84 between the baffle structure 76 and the interior surface portions of the tubular member 36. The gases freely flow into the open ends of the heat trap structures 68 and permeate the sidewalls of the latter between the open end thereof and the deflection plates 70. Deflection plates 70 function to restrict the flow of gases toward the enclosed end of the heat trap structure 68 and thereby tend to evenly distribute the flow of gas out through the pervious sidewalls of the trap structure 68. As the gas flow exits through the pervious sidewalls of the trap structure 68, the sidewalls are heated by conduction, and by virtue of their mesh-like construction, function to trap and store a substantial portion of the heat energy is then partially transferred by conduction through the plate member 72, thence to the interior surface portions of the tubular member 36. Moreover, the deflection plates 70 are likewise heated by conduction, which heat is also conducted through the plate member 72 to the interior surface portions of the tubular member 36. After leaving the heat trap structure 68, the gases flow through the space 84 transferring heat by conduction to both the baffle structure 76 and interior surface portions of the tubular member 36. By virtue of the substantially gas-tight fit between the base of the baffle structure 76 and the interior surface portions of the tubular member 36 adjacent the outlet opening 40, the gases flowing through the spaces 84 are forced to enter the perforations 86 in the baffle structure 76 into the interior spaces of the latter and are then allowed to exit through the outlet opening 40. It can be appreciated that the gases flowing through the tubular member 36 contacts and heats essentially the entire exterior surface area of the baffle structure 76. A portion of the heat extracted by the baffle structure 76 is conducted toward the base thereof and into the interior surface portions of the tubular member 36 adjacent the outlet opening 40. In addition to such conductive heat transfer, the exterior surface portions of the baffle structure 76 function to radiate heat retained thereby toward and onto the interior surface portions of the tubular member 36 essentially the entire length of the baffle structure 76. Moreover, the exterior surface portions of the baffle structure 76 further function to reflect and radiate the heat stored by the heat trap structures 68 toward and onto the interior surface portions of the tubular member 36. On those occasions when the burner or other combustion device employed within the heating chamber 22 is temporarily turned off from time to time, the heat trap structures 68 function to store a certain quantity of heat within the device 34 to maintain the exterior surface portions of the tubular member 36 at a higher temperature level than otherwise.

From the foregoing description of the construction and operation of the heat recovery device 34, it can be appreciated that the three stage structure of the device 34 encourages heat transfer toward the circumferential perimeter of the tubular member 36 through the device 34 to produce a temperature stratification of the gas along the length of the tubular member 36. It may further be appreciated that each of the stages of the heat transferring structure synergistically cooperate with each other by processes of conduction, convection, and radiation to maximize the extraction and transfer of heat from the gases to the interior surface portions of the tubular member 36 for subsequent conduction to the exterior surface areas of the latter, yet the device 34 is especially compact and may be disposed within a stretch of a flue pipe having an ordinary cross-sectional area.

INDUSTRIAL APPLICABILITY

The construction and method of use of the invention have been made amply clear by the foregoing description. It is apparent that although the heat recovery device is disclosed herein as being suitable for use with a furnace system employing a return air duct, the invention has clear applicability to many other operations wherein it is desired to transfer heat from a first flowing medium of one temperature to a second medium of a second temperature. Moreover, it is to be further noted that although the preferred form of the device is disclosed herein as being especially adapted for use in conjunction with flue pipes having a circular cross section, the invention is also well adapted for use with flue pipes having various other cross section configurations. Finally it should be observed that although the three stage structure is employed in connection with the preferred form of the device, each stage may be individually employed in particular applications with acceptable results.

From the foregoing, it is clear that the invention provides a novel device for recovering a substantial portion of the heat from a flue pipe in a furnace system which is especially effective in operation but yet which is particularly compact in size and may be easily installed in existing furnace systems. Thus, it will be observed that the device disclosed herein not only provides for the reliable accomplishment of the invention, but does so in a particularly simple and economical manner. It is recognized, of course, that those skilled in the art may make various modifications or additions to the preferred embodiment chosen to illustrate the invention without departing from the gist and essence of the present contribution to the art. Accordingly, it is to be understood that the protection sought and to be afforded hereby should be deemed to extend to the subject matter claimed and all equivalents thereof fairly within the scope of the invention.

I claim:

1. For use with an air heating furnace of the type having a cool air intake section and a flue for conveying hot gases away from said furnace, apparatus for transferring heat from said gases to the cool air flowing into said furnace through said intake section of the latter, including:

a tubular member formed of a heat conductive material and provided with gas inlet and outlet openings at respective opposite extremities thereof,

said member being adapted to be coupled with said flue with said openings of the latter interposed between stretches of said flue,

said member having exterior surface areas thereof disposed within said intake section and in heat exchanging relationship to said cool air flowing through said section;

first means disposed within said member adjacent said surface areas of the latter for transferring heat from said gases to said surface areas of said member,

said first means including structure extending into the flowpath of said gases through said member between said openings in the latter, and being opera-

tive to deflect at least a portion of the flow of said gases through said member toward and into heat exchanging contact with interior surface portions of said member, but being operative to allow the remaining portion of said gases to flow thereby;

second means disposed within said member adjacent said surface areas of the latter for storing and transferring heat from said gases to said surface areas of said member,

said second means including a gas pervious, heat conducting material generally positioned within said flowpath and allowing said gases to pass freely therethrough, portions of said material being in heat conducting contact with said interior surface portions of said member; and

third means disposed within said member adjacent said surface areas of the latter for transferring heat from said gases to said surface areas of said member,

said third means including elongate baffle means extending longitudinally within said member and having parts thereof in heat conducting contact with said interior surface portions of said member,

said baffle means presenting a substantial obstruction to the flow of said gas through said member for reducing the volume rate at which said gases flow through said member.

2. The invention of claim 1, wherein said first heat transferring means is disposed adjacent said inlet opening of said member and said third heat transferring means is disposed adjacent said outlet opening of said member.

3. The invention of claim 2, wherein:

said tubular member has a circular cross section,

said first means further includes a cylindrically shaped shell circumferentially spaced from said interior surface portions of said member and extending longitudinally, essentially coaxial with said member,

said deflection structure associated with said first means includes a first set of deflection elements circumferentially spaced around the interior of said member and extending from the latter toward the longitudinal axis of said shell into said flowpath of said gases, and

said deflection elements communicate with said shell and deflect at least a part of said portion of said gases to the circumferential space between said shell and said interior surface portions of said member.

4. The invention of claim 3, wherein said deflection structure further includes a second set of deflection elements spaced along the longitudinal axis of said shell from said first set thereof, and circumferentially staggered with respect to the latter, said second set being circumferentially spaced around said interior of said member and functioning to deflect the remaining part of said portion of said gases into said circumferential space.

5. The invention of claim 3, wherein said first heat transferring means further includes:

partitioning structure extending across and partially enclosing one extremity of said shell distal from said inlet opening,

said partitioning structure including a plurality of perforations therein, said perforations functioning to limit the flow of said gases from said inlet opening through said one extremity of said shell to said remaining portion thereof.

6. The invention of claim 5 wherein said partitioning structure includes a circular plate and said perforations are circumferentially spaced around said plate and are essentially equally radially spaced from the longitudinal axis of said shell.

7. The invention of claim 2, wherein said pervious material associated with said second heat storing and transferring means comprises:

a wire mesh-like construction formed into a plurality of individual, generally tubularly-shaped, elongate structures having their longitudinal axes generally aligned with the like axis of said tubular member,

said tubularly-shaped structures being circumferentially spaced about said interior surface portions of said tubular member and including portions thereof secured in heat conducting relationship to said tubular member,

one extremity of said tubularly-shaped structures adjacent said inlet opening being essentially open to receive said gases therethrough,

said tubular spaced structures functioning to extract heat from said gases for transfer to said tubular member.

8. The invention of claim 7, wherein the opposite extremity of each of said tubular structures is closed whereby to force said gases entering through said open extremity thereof to flow out through the openings in the pervious sidewalls of said structure defined by said mesh-like construction, and there is further provided structure mounted generally within each of said tubularly shaped structures for limiting the flow of said gases through said sidewalls between said flow limiting structure and said closed opposite extremity of said tubularly shaped structure.

9. The invention of claim 7, wherein:

said baffle means is essentially hollow and includes longitudinally extending, inclined sidewalls defining a cavity in one extremity of said baffle means adjacent said gas outlet opening in said tubular member,

said sidewalls forming a tapering structure and enclosing the opposite extremity of said baffle means.

10. The invention of claim 9, wherein:

said tapering structure is generally conical in shape, and

the base end of said conically shaped structure is disposed adjacent said gas outlet opening and includes said cavity therein,

essentially the entire circumference of said base end contacting said interior surface portions of said member and preventing flow of said gases therebetween,

said conically shaped structure being provided with a plurality of apertures in said sidewalls thereof placing said gas outlet opening in gas flow communication with said gas inlet opening.

11. Apparatus for removing heat from a flowable medium, comprising:

heat conductive tubular structure for confining the flow of said medium therethrough,

heat conductive transferring means mounted within said tubular structure and in heat transferring relationship to the latter,

said transferring means being disposed within the flowpath of said medium within said tubular structure.

said transferring means and said tubular structure cooperatively operating on said medium to remove a portion of the heat in said medium from the latter,

said transferring means including a first, a second, and a third stage for removing heat from said medium,

said first stage including means for deflecting a portion of said medium toward and into heat transferring contact with interior surface portions of said tubular structure,

said second stage being disposed downstream in the flow of said medium from said first stage,

said second stage including heat trap means comprising heat conducting material pervious to said medium and allowing said medium to pass freely therethrough,

said trap means being conductively coupled with said tubular structure and being operative for conducting heat in said medium away from the latter into said tubular structure as said medium passes through said material,

said trap means including a plurality of elongate members having their longitudinal axes generally aligned with the like axis of said tubular structure,

said third stage being disposed downstream in the flow of said medium from said first stage,

said third stage including regulating means extending across the path of flow of said medium and functioning to reduce the rate of flow of said medium through said tubular structure,

said regulating means including elongate, tapering structure having the sidewalls thereof inclined to the flow of said medium and conductively coupled with said tubular structure,

said trap means being interposed in heat reflecting relationship between said sidewalls of said tapering structure and said interior surface portions of said tubular structure.

12. The invention of claim 11, wherein:

said deflecting means comprises a plurality of deflecting elements, and

said first stage further includes a sleeve member having the sidewalls thereof spaced from said interior surface portions of said member for confining the flow of said portion of said medium between said sleeve member and said interior surface portions of said tubular structure,

said first stage further including means for directing the flow of said medium toward and into said pervious material associated with said second stage,

said tapering structure being essentially hollow and generally conical in shape, and having the base end thereof essentially open,

said tapering structure including a plurality of perforations in said sidewalls thereof placing spaces within said tubular structure upstream in said flow from said third stage in medium flow communication with spaces in said tubular structure downstream in said flow from said third stage.

* * * * *

United States Patent [19]

Gruber et al.

[11] 4,203,415

[45] May 20, 1980

[54] HEAT EXCHANGER

[76] Inventors: Thomas J. Gruber, 7700 Fairmount Rd., Novelty, Ohio 44072; Harris W. Smith, 16839 Savage Rd., Bainbridge, Chagrin Falls, Ohio 44022

[21] Appl. No.: 896,881

[22] Filed: Apr. 17, 1978

[51] Int. Cl.[2] F24H 3/02

[52] U.S. Cl. ..... 126/110 R

[58] Field of Search ..... 126/110 R, 110 AA, 110 D, 126/110 E

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Primary Examiner—Albert W. Davis

Attorney, Agent, or Firm—Fay & Sharpe

[57] ABSTRACT

A new and improved heat exchanger of the cross over type which facilitates heating an air flow directed either axially or transversely of the heat exchanger itself. The exchanger is comprised of an inner shell which defines a heating chamber for a fuel burner and an outer shell which is coaxially received over the inner shell in a radially spaced apart relationship therewith. The generally annular area between the inner and outer shells defines an air flow passage with the air flow to be heated being primarily conveyed through this air flow passage. A cross over member extends across the air flow passage to provide gaseous flow communication between the heating chamber and a gaseous flow chamber associated with the outer shell. This arrangement facilitates continuously conveying the products of combustion outwardly from the heating chamber. A thin air deflector surrounds the cross over member and is disposed in the air flow passage generally parallel to both the inner and outer shells. This air deflector acts as a heat sink and promotes laminar air flow around the cross over member. The outer shell also includes a pair of oppositely disposed elongated slot-like openings disposed in registry with at least a portion of the inner shell. The heat exchanger may be advantageously mounted either axially or transversely of an air flow to be heated. When axially mounted, at least a portion of the air flow is axially through the air flow passage and when transversely mounted, at least a portion of the air flow is transversely through the air flow passage between the slot-like openings in the outer shell.

13 Claims, 6 Drawing Figures

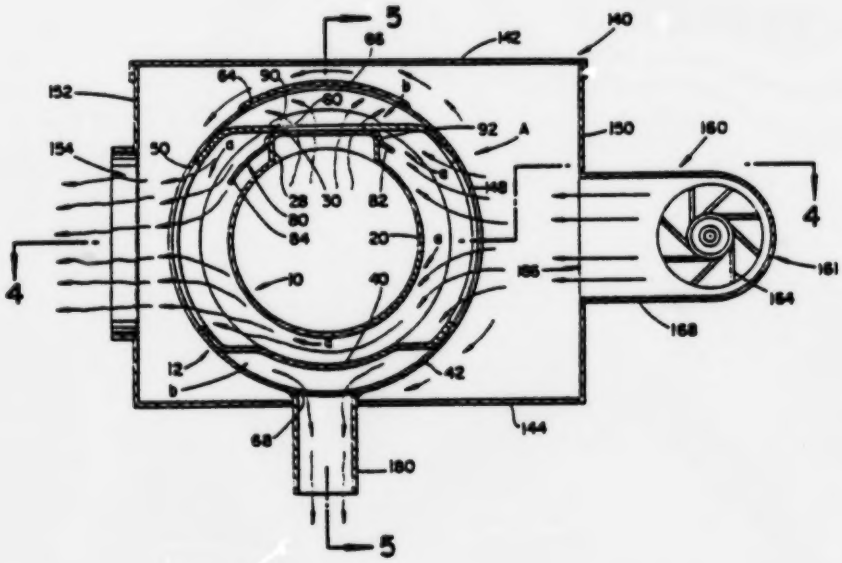

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FIG. 1

FIG. 2

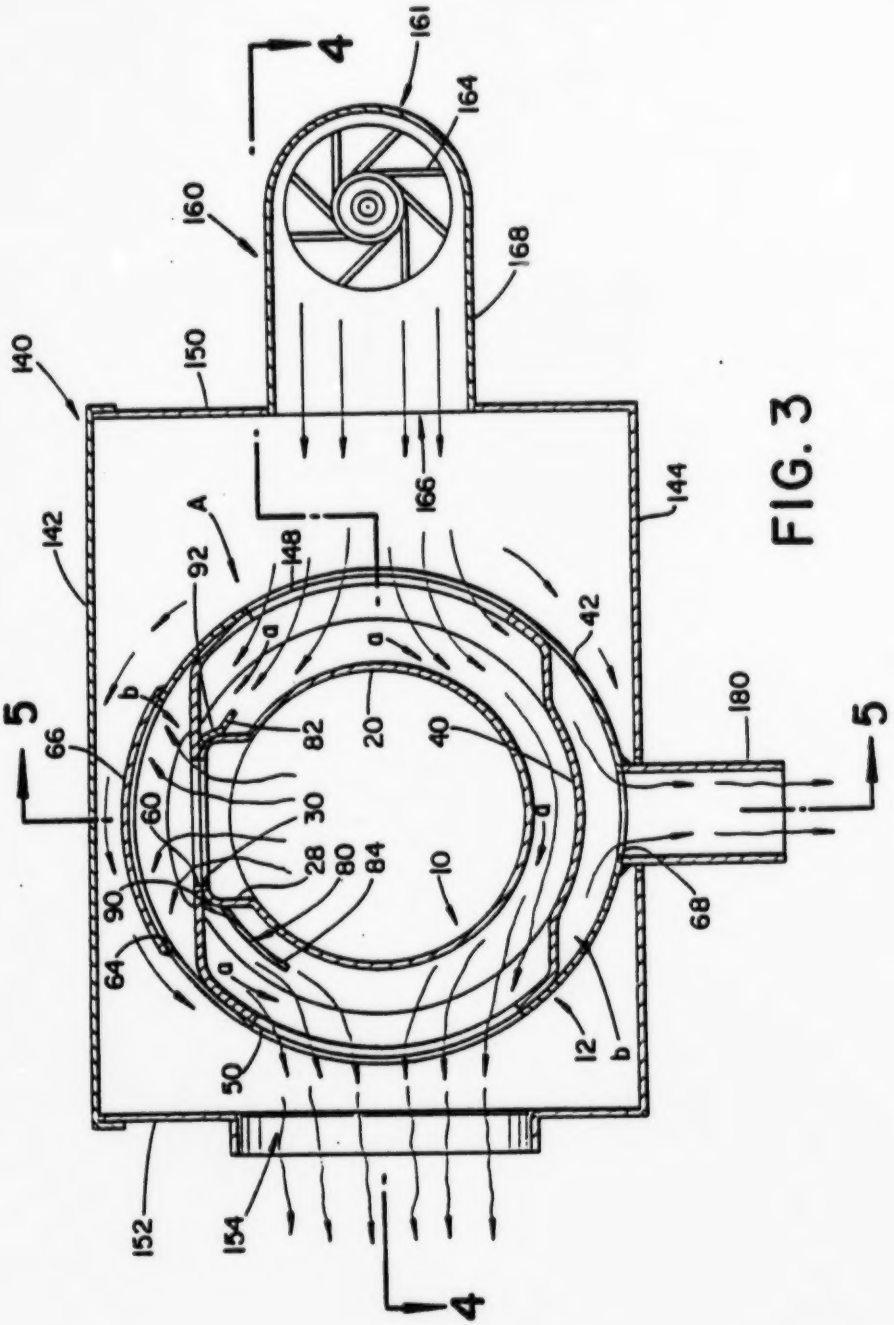
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FIG. 3

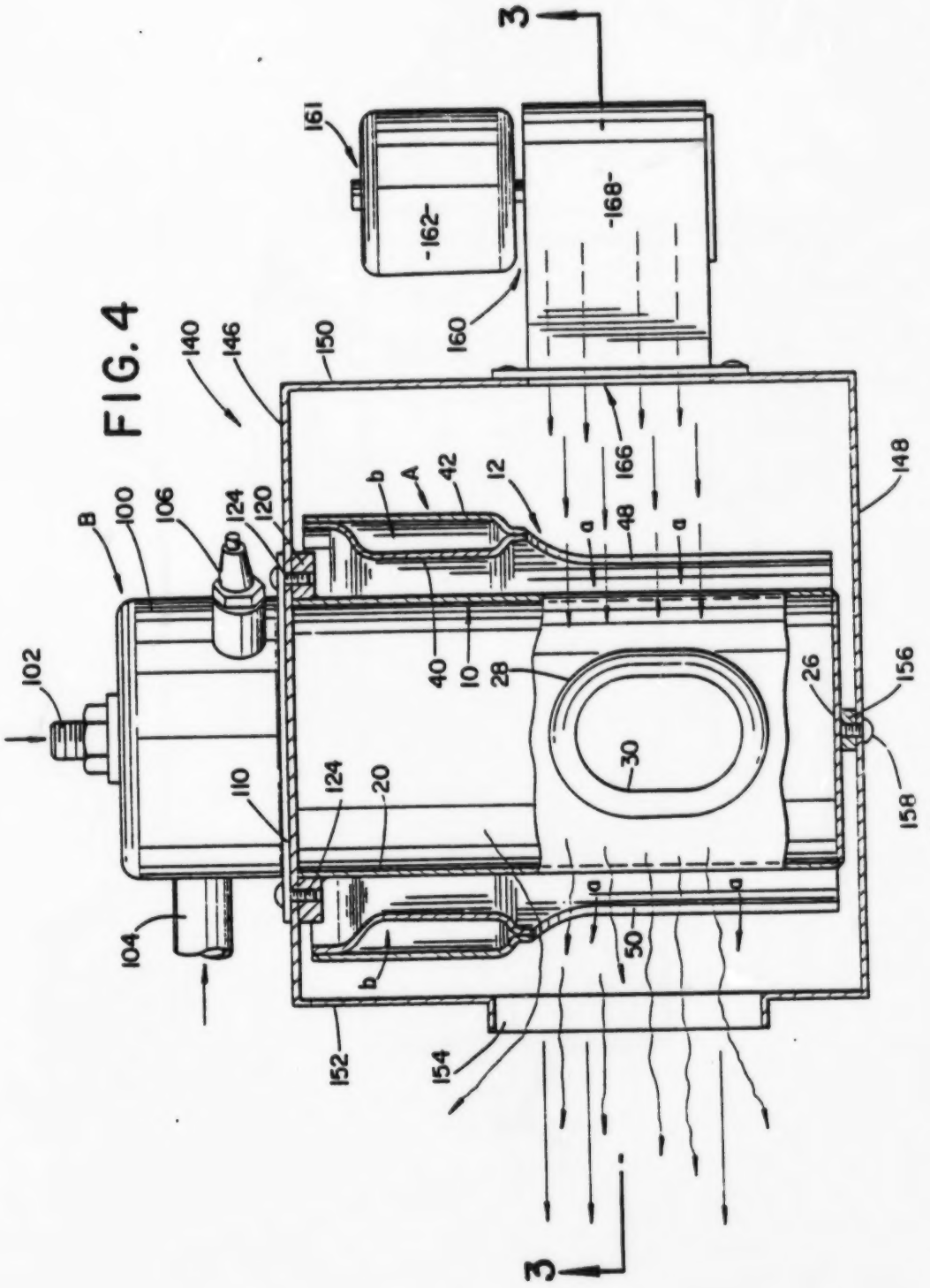
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FIG. 4

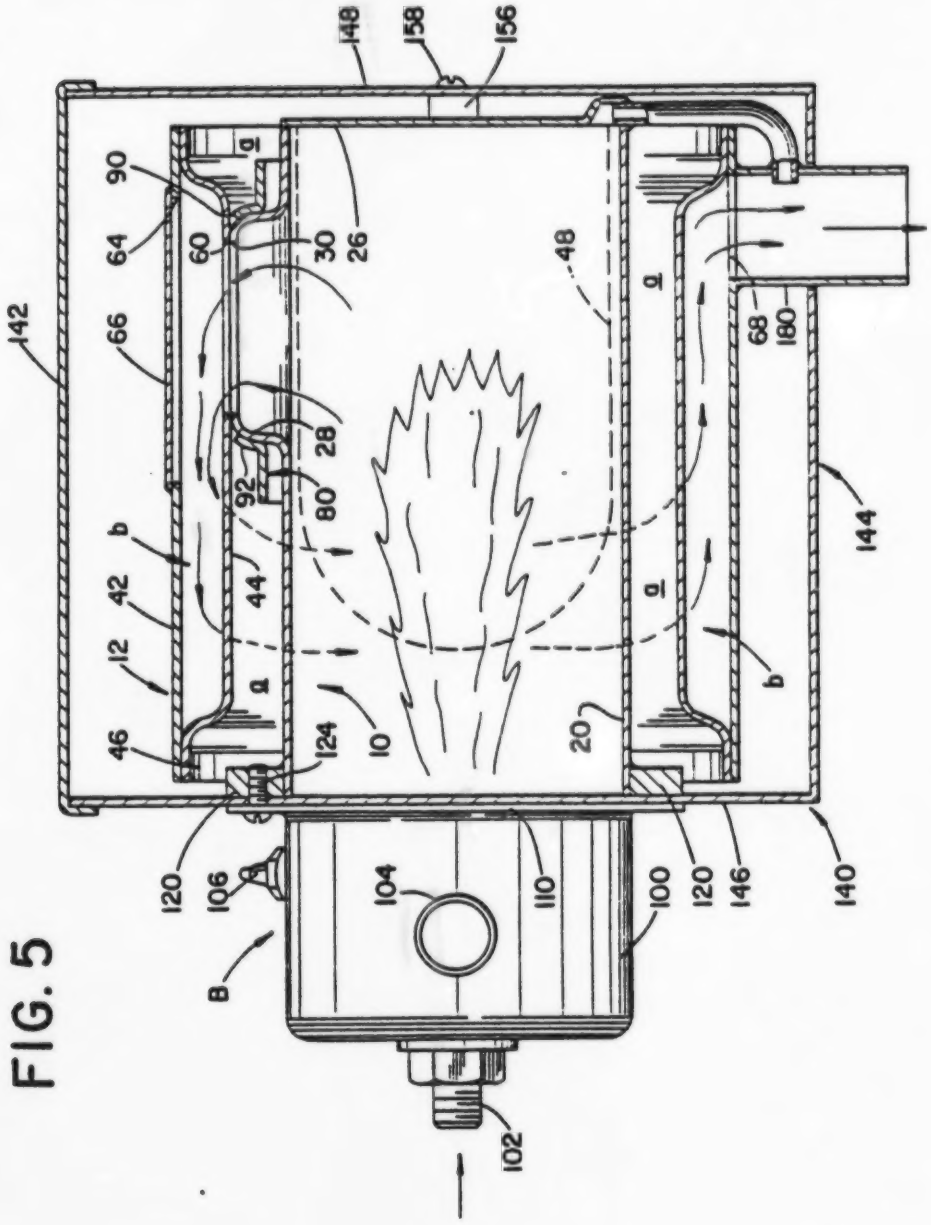
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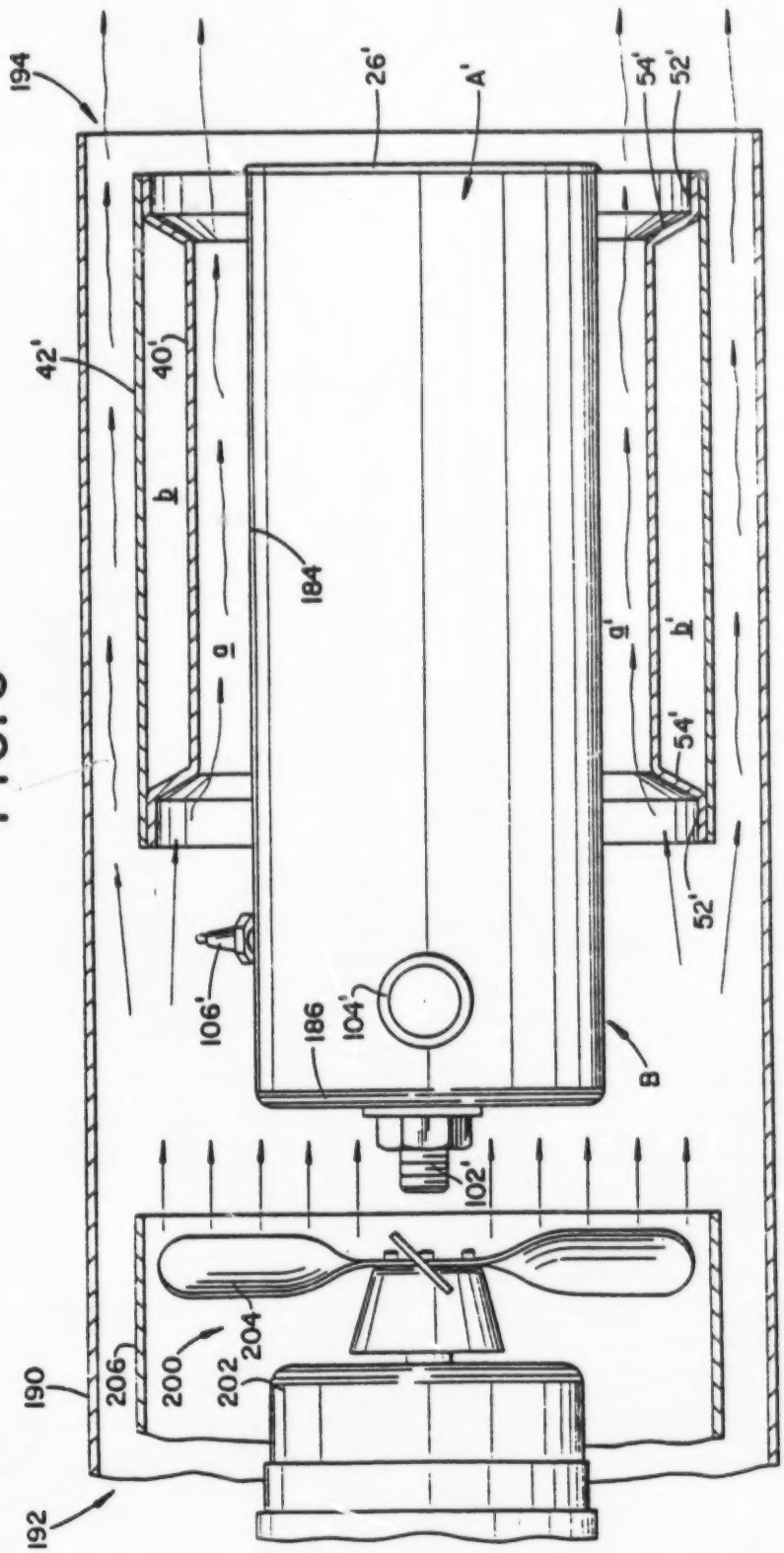
FIG. 5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g
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102'
204
200
202
206
190
192

FIG. 6

HEAT EXCHANGER

BACKGROUND OF THE INVENTION

This invention pertains to the art of heat exchangers and more particularly to heat exchangers used for heating a flow of air passed in close communication therewith.

The invention is particularly applicable to a heat exchanger of the cross-over type which may be either axially or transversely mounted relative to an air flow and will be described with particular reference thereto. However, it will be appreciated by those skilled in the art that the invention has broader applications and uses.

Heat exchangers of the cross-over type employed for heating a continuous flow of air passed in close proximity thereto have been known in the art. Generally, this type of heat exchanger design includes some type of inner shell which defines a heating chamber and which is adapted to receive a burner in operative association therewith for causing combustion of a continuous fuel supply therein. Surrounding this inner shell and radially spaced therefrom is an outer shell with the space between the inner and outer shells defining an air flow passage. The outer shell includes one of various means for defining a gaseous flow passage or chamber thereover and which passage or chamber is placed in gaseous flow communication with the heating chamber through one or more cross over members or areas. This arrangement allows the products of combustion generated in the heating chamber to be conveyed therefrom by means of the cross over member or area into the gaseous flow passage or chamber and then exhausted outwardly of the heat exchanger structure. An air flow to be heated is continuously directed by fan means or the like through at least the air flow passage into close proximity with the inner and outer shells for heating and subsequent dispersement for heating a room, other enclosure or the like.

However, prior heat exchangers of this general type have had several drawbacks. First, the designs have been such that they were adapted to provide heating for an air flow passed in only a single direction relative to the heat exchanger. Typically, this direction was axially of the exchanger and air flow passage but there have been some exchanger designs adapted for transverse mounting relative to an air flow. However, no prior heat exchanger designs of this general type have been developed which would successfully facilitate alternative mounting of the heat exchanger either axially or transversely of an air flow. Thus, for different heat exchanger applications, i.e., where it was necessary or desirable for some reason to mount a heat exchanger in one direction or another relative to an air flow, different heat exchanger designs and embodiments had to be provided.

Another problem with prior heat exchanger designs of this general type has been in the conveyance of the products of combustion from the heating chamber, through the cross over member or area and then through the passage or chamber in the outer shell to an exhaust area. In view of the fact that the products of combustion are hot and provide an additional vehicle for heating an air flow, it is desirable to provide a sinuous travel path for the products of combustion along the passage or chamber in the outer shell to achieve a more efficient air flow heating arrangement. However, prior art heat exchangers have not fully capitalized on this desirable arrangement and, quite often, simply provided a simple or direct gaseous flow path to the exhaust area.

Still another problem area with prior heat exchangers of this general type has been in accommodating flow through the air flow passage around the cross over member or area itself. Because this cross over member or area is oftentimes located directly in the air flow path and conveys the very hot products of combustion from the heating chamber to the gaseous flow passage or chamber in the outer shell, a substantial heat build-up is present at the cross over area. Because of this heat build-up, it was necessary to utilize special and costly heat resistant metals or materials for constructing the cross over member in order to avoid breakdown of the heat exchanger at this area. Moreover, the cross over member comprises an impediment to smooth air flow along the air flow passage and reduces the overall heat exchanger efficiency. Thus, the amount of fuel and electrical power required to achieve a desired heat exchanger rating is higher than it might otherwise be.

The subject invention contemplates new and improved apparatus which overcomes all of the above referred to problems and which provides a new heat exchanger structure of the cross over type which is simple in design, economical to manufacture, provides increased heating efficiencies by fully utilizing the heated products of combustion, provides an improved air flow around and a heat sink at the cross over member or area and which is readily adapted to mounting either axially or transversely of an air flow to be heated.

BRIEF DESCRIPTION OF THE INVENTION

In accordance with the present invention, there is provided a new heat exchanger construction adapted to be mounted in one of an axial and transverse disposition relative to and in the path of a flow of air which is to be heated. The heat exchanger is comprised of an elongated hollow inner shell having a side wall, a closed inner end and an outer end and defining a heating chamber therein. The outer end is adapted to have a fuel burner disposed in operative association therewith for causing combustion of a continuous fuel supply introduced into the burner and passing generally axially through the heating chamber toward the closed inner end. An elongated outer shell is disposed generally coaxially around the inner shell and radially spaced therefrom to define an air flow passage. The outer shell has first and second ends and spaced apart inner and outer walls with the inner and outer walls being sealed at the terminal edges thereof to define a gaseous flow chamber therebetween for the products of combustion generated in the heating chamber. As a result of this structure, at least an air flow passing into the air flow passage at a first temperature may be heated to a second temperature elevated from the first by transfer thereto of heat generated in the heating chamber and in the gaseous flow chamber. Means for defining a gaseous flow channel or cross over area communicates between the heating chamber from adjacent the closed inner end thereof and the gaseous flow chamber for conveying the products of combustion generated in the heating chamber across the air flow passage and into the gaseous flow chamber. A gaseous flow outlet communicates with the gaseous flow chamber at a position thereon spaced from the channel defining means or cross over area for conveying or exhausting products of combustion outwardly from the gaseous flow chamber. The

outer shell also includes at least a pair of openings extending generally transversely through the inner and outer walls thereof in registry with at least a portion of the inner shell. With this construction, the heat exchanger may be mounted axially of an air flow with the air flow being heated from the first to the second temperature as it flows axially through the air flow passage or may be mounted transversely of an air flow with the air flow being heated from the first to the second temperature as it flows generally transversely through the air flow passage between the pair of openings in the outer shell.

In accordance with another aspect of the present invention, the channel defining means and outlet are disposed in communication with the gaseous flow chamber on opposite sides of the outer shell. The at least a pair of openings in the outer shell comprise slot-like areas generally oppositely disposed from each other intermediate the channel defining means and the outlet. With this particular arrangement, the products of combustion passing from the heating chamber into the gaseous flow chamber through the channel defining means may follow a serpentine flow path through the gaseous flow chamber to the outlet or exhaust area.

In accordance with another aspect of the present invention, the heat exchanger includes a thin air flow deflector disposed in the air flow passage intermediate the inner and outer shells and surrounds the channel defining means. This deflector extends generally parallel to the inner and outer shells and at least acts as means for efficiently conveying the air flow axially or transversely of the heat exchanger through the air flow passage around the channel defining means.

In accordance with still another aspect of the present invention, an improvement is provided for a heat exchanger of the type having a generally cylindrical inner shell defining a heating chamber and an outer generally cylindrical shell coaxially disposed relative to and radially spaced from the inner shell thereby defining generally annular air flow passage between the inner and outer shells for heating a flow of air passing therethrough with the outer shell including means for conveying products of combustion generated in the heating chamber by a fuel burner and with the heating chamber and conveying means being placed in gaseous flow communication with each other across the air flow passage by a cross over member or area. The improvement comprises a thin air flow deflector surrounding the cross over member or area in the air flow passage intermediate the inner and outer shells for at least efficiently conveying air flow in the air flow passage around the cross over member or area.

In accordance with a still further aspect of the present invention, there is provided an improvement for a heat exchanger of the type employed to heat a continuous flow of air passing between the outer side wall of an inner shell defining a heating chamber and the inner side wall of an outer shell which surrounds the heating chamber wherein heat is continuously transferred at least from the outer side wall of the heating chamber to the continuous flow of air for subsequent distribution for heating enclosed areas and the like. The improvement is designed to permit mounting of the heat exchanger either axially or transversely of the air flow and comprises the inner and outer shells being generally coaxially disposed relative to each other with the outer shell inner wall spaced from the inner shell outer wall so as to define an air flow passage therebetween over the axial lengths thereof whereby the flow of air may be selectively directed axially of the heat exchanger through the air flow passage for heating purposes. Further, the outer shell includes at least a pair of openings passing therethrough in registry with at least a portion of the inner shell whereby the flow of air may be selectively directed transversely of the heat exchanger through the air flow passage between the openings.

The principal object of the present invention is the provision of a new and improved heat exchanger of the cross over type which is simple in design and construction.

Another object of the present invention is the provision of a heat exchanger of the cross over type which is efficient in operation.

Still another object of the present invention is the provision of a heat exchanger of the cross over type which provides means for efficient air flow around the cross over member or area.

Yet a further object of the present invention is a heat exchanger of the cross over type wherein the exchanger may be selectively mounted either axially or transversely of an air flow which is to be heated.

DESCRIPTION OF THE DRAWINGS

The invention may take physical form in certain parts and arrangements of parts, preferred embodiments of which will be described in detail in the specification and illustrated in the accompanying drawings which form a part hereof and wherein:

FIG. 1 is an exploded perspective view of the subject heat exchanger with a portion thereof broken away for ease of illustration;

FIG. 2 is a perspective view showing the heat exchanger of FIG. 1 in the assembled condition;

FIGS. 3–5 show various cross-sectional views of the heat exchanger when mounted to accommodate an air flow transverse of the exchanger; and,

FIG. 6 shows the heat exchanger mounted to accommodate air flow axially thereof and also shows a slightly modified structural arrangement for the heat exchanger inner shell and burner body.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings wherein the showings are for purposes of illustrating the preferred embodiment of the invention only and not for purposes of limiting same, the FIGURES show a heat exchanger A which receives a fuel burner B in close spaced operative association therewith.

With particular reference to FIGS. 1 and 2, heat exchanger A is basically comprised of a cylindrical inner shell generally designated 10, a cylindrical outer shell generally designated 12 which is coaxially received over the inner shell and a cross over member with deflector generally designated 14. These components are constructed from high temperature resistant sheet steel in order to accommodate the high operating temperatures and further, are of welded construction as will be further described and appreciated hereinafter.

Inner shell 10 is comprised of an elongated cylindrical side wall 20 having open inner and outer ends 22,24. A cover or end plate 26 is rigidly affixed to side wall 20 at inner end 22 so as to close that end and provide the inner shell with a generally cup-shaped configuration. The inside of inner shell 10 defines a heating chamber which is used in a manner to be hereinafter described.

Formed directly into side wall 20 is an outwardly extending boss area 28 terminating in a slightly curved top wall area 29 (FIG. 5) defining a generally oval opening 30 therethrough communicating with the heating chamber itself. While a number of manufacturing techniques may be employed for manufacturing side wall 20, it is constructed from a flat piece of sheet steel which is rolled or formed into the cylindrical configuration and then welded along a longitudinally extending seam area. Boss area 28 may be conveniently provided in the side wall by conventional metal working techniques such as drawing and its outward extension from side wall 20 is such that it defines a cross over area and receives an air deflector in a manner to be described hereinafter.

Outer shell 12 is comprised of an inner cylindrical member 40 defining an inner wall and an outer cylindrical member 42 defining an outer wall. Members 40,42 are coaxially and coextensively disposed relative to each other and have a first end 44 and a second end 46. As will be particularly noted from FIG. 1, the inner and outer cylindrical members are diametrically dimensioned such that there is an annular space between them when they are coaxially disposed relative to each other. Moreover, the inner diameter of outer shell 12 is greater than the outer diameter of inner shell 10 for defining an air flow passage which will be described in greater detail. Extending inwardly from first end 44 longitudinally toward second end 46 are a pair of generally U-shaped slot areas 48,50 which pass through both cylindrical members 40,42. While a number of different types and styles of slots could be employed, the generally U-shaped configuration is preferred. Moreover, these slots are substantially identical to each other in size and shape and preferably extend from first end 44 toward second end 46 over at least slightly more than one half of the length of outer shell 12. Also, each of the opposed slots 48,50 has a width slightly under one fourth of the circumference of outer shell 12. These length and width dimensions may be varied as desired or necessary without in any way affecting the scope and intent of the invention.

As will be particularly noted from FIGS. 1 and 6, inner cylindrical member 40 is formed so as to include flange areas generally designated 52 adjacent the terminal edges thereof forming border areas which are rigidly affixed to the associated terminal edges of outer cylindrical member 42. With this contruction, outer shell 12 comprises a basically hollow structure with the two cylindrical members 40,42 connected together adjacent the mutually associated terminal edges thereof. Generally outwardly extending transition areas 54 connect the main portion of cylindrical member 40 and flange areas 52 for defining end walls in the outer shell 12 structure. Both inner and outer cylindrical members 40,42 are constructed from sheet metal material, formed into a cylindrical configuration and then joined along a longitudinal seam. Moreover, flange and transition areas 52,54 of the inner cylindrical member may be formed by conventional metal working techniques. The longitudinal seams and interconnection between members 40,42 at flange areas 52 may be advantageously performed by conventional welding techniques. Disposed intermediate slots 48,50 in cylindrical member 40 is a generally oval opening 60. Opening 60 is generally compatible in size and shape with opening 30 in boss 28 of inner shell 10.

Oppositely disposed from opening 60 in outer cylindrical member 42 is slightly larger opening 64. This opening facilitates assembly of the overall heat exchanger structure as will become more readily apparent hereinafter. A cover 66 is provided for opening 64 and may be conveniently welded thereto by conventional welding techniques once the necessary assembly process through the opening has been completed. An exhaust opening 68 (FIGS. 3 and 5) is oppositely disposed from opening 64 in cylindrical member 42. Use of this exhaust opening will be discussed in greater detail hereinafter.

With continued reference to FIG. 1, an air deflector 80 provides improved air flow characteristics during use of the heat exchanger and includes a leading edge 82, a trailing edge 84 and opposed side edges 86,88. A generally centrally located oval opening 90 passes through the air deflector and is surrounded by a mounting flange 92. Opening 90 and mounting flange 92 are dimensioned so as to be closely received over boss 28 for practical mounting and for operational reasons which will become apparent hereinafter. In addition to providing improved air flow characteristics, deflector 80 acts as a heat sink in a manner to be described. Moreover, it is also possible to construct deflector 80 to eliminate mounting flange 92 and simply dimension opening 90 to be closely received over boss 28.

Still with particular reference to FIG. 1, fuel burner B is adapted for mounting in an operative relationship with the exchanger at open outer end 24 of inner shell 10. The burner has a burner body 100 and includes fuel and air inlets 102 and 104 (FIGS. 4 and 5). An ignitor generally designated 106 penetrates the side of the burner for igniting the mixture of fuel and air received through inlets 102,104 into a combustion nozzle 108. As the specifics of construction, fueling and operation of such burners are deemed known in the art, further elaboration thereon is deemed unnecessary for a full and complete understanding of the subject invention. The burner in and of itself, however, does not comprise a part of the heat exchanger per se but rather, is necessary for operation and use thereof. Body 100 of burner B includes an outwardly disposed flange 110 (FIGS. 4 and 5) which may be conveniently used to mount the burner to a mounting flange generally designated 120 which itself is mounted in a manner described hereinafter. This mounting flange has a plurality of mounting openings 122 spaced therearound to receive conventional mechanical fasteners 124 for the actual mounting operation as is best shown, for example, in FIGS. 4 and 5. It is also possible to form burner body 110 integral with inner shell 10 in a manner described hereinafter and shown with reference to FIG. 6.

With particular reference to both FIGS. 1 and 2, the above described and discussed individual components are mounted relative to each other in a manner so as to form the final heat exchanger structure shown in FIG. 2. More particularly, outer shell 12 is coaxially disposed over inner shell 10 and since the inner diameter of outer shell 12 is greater than the outer diameter of inner shell 10, an annular air flow passage generally designated a is formed between the two shells. The inner and outer shells are positioned such that first end 44 of the outer shell is generally coextensive with the inner end of the inner shell as defined by cover or end plate 26. Second end 46 of the outer shell is generally coextensive with the outer end 24 of the inner shell although, preferably, outer end 24 extends slightly outwardly of second end 46. In this arrangement, U-shaped slots 48,50 are disposed in registry with a portion of the inner shell side

wall 20. This is deemed important from an operational standpoint for reasons which will be more fully described. As used hereinafter, axial flow is deemed to be generally longitudinally of the heat exchanger between ends 46,44 and transverse air flow is deemed to be generally normal thereto.

The only place of interconnection between inner and outer shells 10,12 is at boss areas 28. More particularly, air deflector 80 is received over the boss and rigidly affixed thereto with the two shells being rigidly joined together about openings 30,60. FIG. 5 best shows this particular arrangement where bosses 28 receives the air deflector through opening 90 with mounting flange 92 closely positioned about the boss adjacent the outer end thereof and with openings 30,60 placed in registry with each other. A strong joint is necessary at this area and conventional welding techniques may be advantageously employed. The access provided through opening 64 is considered to be desirable for this particular assembly operation. The joint or interconnection between the inner and outer shells is such that the heating chamber defined on the inside of inner shell 10 is placed into gaseous flow communication with a gaseous flow chamber b defined between inner cylindrical members 40,42 and which chamber is also placed into flow communication with exhaust opening 68 as described hereinabove. The combination of boss 28 with its opening 30 and opening 60 in inner cylindrical member 40 passing therethrough define a cross over member or area. However, other arrangements for achieving a cross over member or area could also be utilized. Mounting flange 120 may be conveniently mounted to the heat exchanger structure at open outer end 24 of inner shell 10 by, for example, welding the flange thereto.

In using the concepts of the subject invention, particular attention is invited to FIGS. 3–5 which show the heat exchanger mounted transversely to a flow of air which is to be heated. In these FIGURES, and merely for illustrating one type of heat exchanger installation, the heat exchanger and burner are shown as being operably disposed in a housing design generally designated 140. The housing is configured to include a cover or top wall 142, a bottom wall 144, spaced apart side walls 146,148 and spaced apart end walls 150,152. An outlet generally designated 154 is included in end wall 152 to permit the heated air flow to pass into the area which itself is to be heated thereby. The housing itself may be manufactured from many known and conventional materials and the specific design of the housing may be varied as necessary and/or desirable for accommodating different installation needs.

In the example shown, heat exchanger A and burner B are axially disposed between side walls 146,148 in a manner such that the heat exchanger is fully received in the housing and the burner is disposed outside the housing. To accommodate this arrangement, side wall 146 includes a generally circular opening therethrough. The heat exchanger and burner may be affixed in place relative to the housing by any number of convenient and alternative means. However, in the example housing shown, a portion of side wall 146 adjacent the circular opening is received between burner flange 110 and mounting flange 120 with mechanical fasteners 124 passing therethrough. A spacer or bracket generally designated 156 is interposed between cover or end plate 26 of inner shell 10 and the inside of side wall 148. A mechanical fastener 158 is shown as passing through side bottom wall 148 and into the spacer or bracket to retain the end of the heat exchanger in position. It should also be noted that top and bottom walls 142,144 are fairly closely spaced to outer shell 12 for promoting air flow through the heat exchanger rather than around it. The specific mounting arrangement shown and just described is exemplary for accommodating transverse air flow and, again, may be modified or varied as deemed necessary for a particular application without in any way departing from the intent or scope of the invention.

A fan assembly generally designated 160 (FIGS. 3 and 4) is disposed adjacent end wall 150 for providing a continuous air flow transversely across the heat exchanger. The fan assembly is housed in an assembly housing 161 extending outwardly of side wall 148 and the assembly itself includes a motor generally designated 162 which drives a fan 164 for creating an air flow outwardly from an outlet 166 in the fan housing 168. The specifics of the fan assembly do not form a part of the present invention and are not, therefore, described in any greater detail therein. It will be appreciated by those skilled in the art that many other fan or alternative air flow supply arrangements could also be advantageously employed in practicing the concepts of the subject invention. To complete the general arrangement shown in FIGS. 3–5, an exhaust tube generally designated 180 communicates between exhaust opening 68 in outer cylindrical member 42 of outer shell 12 and bottom wall 144 of housing 140.

In using the structural arrangement of FIGS. 3–5 for heating an air flow, burner B is, of course, provided with appropriate air and fuel supplies through inlets 102,104 as is conventional. Ignitor 106 is connected to an appropriate electrical source so that the combustible fuel mixture passing through nozzle 108 may be continuously ignited. With such a conventional fuel burner hook-up, the burner is placed into operation and the resulting flame as generally schematically shown in FIG. 5 causes a heating action in the heating chamber defined within inner shell 10. Heat thus generated is transferred to the walls of the inner shell itself and the products of combustion from the fuel burner are continuously passed from the heating chamber through openings 30,60 into the gaseous flow chamber.

With particular reference to FIG. 5, the path taken by the gaseous products of combustion is shown by the arrowed lines passing from the heating chamber and into gaseous flow chamber b. Once in gaseous flow chamber b, the products of combustion flow upwardly toward second end 46 of outer shell 12, then circumferentially around the upper portion of the shell, then downwardly toward and out of the chamber at exhaust opening 68 through exhaust tube 180. Thus, the products of combustion take a serpentine path from the entrance area into gaseous flow chamber b at the cross over member or area to the exhaust opening therefrom. In addition to the above description with reference to FIG. 5, the overall serpentine nature for the flow of gaseous products of combustion through chamber b can be best appreciated by viewing the partially cross sectioned outer shell shown in FIG. 1. In FIG. 1 and from the point of entry of the products of combustion into the gaseous flow chamber through opening 60, the gaseous products travel upwardly around U-shaped slots 48,50 and then downwardly toward opening 68 for exhaust from the chamber. Because these products of combustion are also quite hot, they will heat both inner and outer cylindrical members 40,42 during their course of

travel through gaseous flow chamber b. Due to the serpentine nature of flow in chamber b, much of the heat in the products of combustion will be effectively transferred to the inner and outer cylindrical members.

Referring now particularly to FIGS. 3 and 4, the arrowed lines originating from fan assembly 160 schematically show the flow of air which is to be heated by being passed into close communication and proximity with the heat exchanger. This air flow is, of course, focused so as to flow transversely of the heat exchanger and is further focused (FIG. 4) so as to flow through slot 48 into air flow passage a defined between inner and outer shells 10,12. From there, the air flows around the inner shell (FIG. 3), then toward and outwardly of passage a at slot 50 and then outwardly of housing 140 at outlet 154 for distribution to heat a room, enclosure or the like.

As the air flow passes through passage a generally transverse of the exchanger, heat is transferred thereby by convection from the outside wall of inner shell 10 and the inside wall of outer shell 12. Thus, when the air flow exits air flow passage a at slot area 50, the temperature thereof is raised from a first ambient temperature as supplied from fan assembly 160 to a second elevated temperature which may then be employed to heat a room, enclosure or the like. Some of the air flow originating at fan assembly 160 will flow around the outside of outer shell 12, i.e., around outer cylindrical member 42. Since the products of combustion heat both inner and outer cylindrical members 40,42, this air will also have been heated to some extent as it flows outwardly of housing 140 through outlet 154.

A particularly advantageous aspect of the present invention is the affect that air deflector 80 has on the air flow passing thereby within air flow passage a. This deflector, best shown in FIGS. 3 and 5 in an operative position, acts to reduce heat build-up at the cross over member or area since through the intimate contact between mounting flange 92 and boss 28, it functions as a fin or heat sink and provides additional surface area for air flow thereover resulting in better heat transfer efficiency. Since the deflector acts to reduce heat buid-up at the cross over member or area, it is not necessary to utilize expensive and extremely high temperature resistant materials for forming the cross over member or area.

Indeed and as noted above, this cross over member or area may be formed from the parent material of shells 10,12 and integral therewith. More importantly, however, the air deflector assists in spreading an acceptable heat gradient over the entire exchanger. In addition to acting as a heat sink, the air deflector improves air flow characteristics across the heat exchanger through passage a so as to minimize turbulence and provide generally laminar air flow on the back side of the deflector adjacent trailing edge 84 thereby increasing the total air flow across the heat exchanger. The particularly advantageous results obtained by using air deflector 80 are in providing a far more efficient heat exchanger structure and adding to the total exposed area available to transfer heat from the cross over member or area. This results in less fuel being used to achieve the final desired heating for an air flow as well as requiring less fuel and electrical power to achieve a desired BTU/hour heat exchanger rating. If the alternative structure for the air deflector which eliminates mounting flange 92 is used, however, the necessary intimate contact between the deflector and boss 28 is not provided to permit it to effectively act as a heat sink.

In addition to installation and use of the heat exchanger in conjunction with an air flow directed generally transversely thereof as shown and described above with reference to FIGS. 3-5, the heat exchanger is readily alternatively adapted to installation and use in an axial air flow environment as shown in FIG. 6. FIG. 6 also shows a slightly modified structural arrangement for the heat exchanger inner shell and the burner body. For ease of illustration and appreciation of this modification, like components are identified by like numerals with a primed (') suffix and new components are identified by new numerals.

In that regard, the heat exchanger inner shell and burner body comprise a single elongated open ended tubular shell 184. Cover or end plate 26' is affixed to one end of this shell and a burner nozzle support plate 186 is affixed to the other end. Fuel nozzle or inlet 102' penetrates plate 186 generally coaxially of shell 184 with air inlet 104' and ignitor 106' radially penetrating the side wall of shell 184 substantially as discussed above with reference to FIGS. 1-5. The connections for inlets 102',104' and ignitor 106' facilitate operation of the burner in a manner as also described above. The remainder of the overall heat exchanger A' construction is as previously described.

In this axial flow embodiment, heat exchanger A' with the integral burner is disposed in an elongated housing generally designated 190 having a lead end 192 and an exit end 194. A fan assembly generally designated 200 includes a motor 202 for driving a fan 204 and which assembly is disposed in a fan housing generally designated 206. During operation of the fan assembly, an air flow is directed from adjacent lead end 192 toward exit end 194 axially of heat exchanger A'. With the heat exchanger and burner operating as hereinabove previously described, a portion of the air flow will pass into air flow passage a' and then pass axially therealong to be heated in the same manner also previously described. In addition, a certain portion of the air flow will pass axially along and in close proximity to the outer side portion of outer cylindrical member 42' and be heated thereby. The heated air flow exiting from exit end 194 of housing 190 may then be utilized for heating a room, enclosure or the like similar to that hereinabove previously described. Moreover, the same advantageous results will be obtained from air deflector 80'. Axial mounting arrangements for the heat exchanger other than that shown in FIG. 6 may be advantageously employed without in any way departing from the intent or scope of the present invention. Moreover, the integrally formed heat exchanger inner shell and burner body shown in this FIGURE could also be advantageously adapted to use in the cross flow environment of FIG. 3-5.

As shown in the FIGURES and described hereinabove in detail, the subject heat exchanger provides an arrangement which may be advantageously employed in heating an air flow directed either axially or transversely of the heat exchanger itself. This structure provides a degree of heat exchanger versatility heretofore not obtainable in prior heat exchanger designs. Because of the overall arrangements of prior designs, they could not be effectively alternatively utilized to heat either axial or transverse air flow. In addition, the subject new heat exchanger provides for improved overall opera-

tional efficiency and means for reducing heat build up at the cross over member or area.

The invention has been described with reference to the preferred embodiment. Obviously, modifications and alterations occur to others upon the reading and understanding of this specification. It is intended to include all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.

Having thus described the invention, it is now claimed:

1. A heat exchanger construction adapted to be mounted in one of an axial and transverse disposition relative to and in the path of a flow of air which is to be heated thereby, said heat exchanger comprising:

an elongated hollow inner shell having a side wall, a closed inner end and an outer end with said inner shell defining a heating chamber therein, said outer end adapted to have a burner disposed in operative association therewith for causing combustion of a continuous fuel supply introduced into said burner and passing generally axially through said chamber toward said closed inner end;

an elongated outer shell disposed generally coaxially around said inner shell and spaced therefrom to define an air flow passage, said outer shell having first and second ends and spaced apart inner and outer walls with said inner and outer walls being sealed together at least at the terminal edges thereof for defining a gaseous flow chamber therebetween for the products of combustion generated in said heating chamber, whereby at least an air flow passing into said air flow passage at a first temperature may be heated to a second temperature elevated from the first by heat transfer thereto of heat generated in said heating chamber and said gaseous flow chamber;

means defining a gaseous flow channel communicating between said heating chamber from adjacent the closed inner end thereof and said gaseous flow chamber adjacent said outer shell first end for conveying products of combustion generated in said heating chamber into said gaseous flow chamber;

a gaseous flow outlet in said outer shell cummunicating with said gaseous flow chamber adjacent said outer shell first end at a position spaced from said channel defining means for conveying products of combustion outwardly from said gaseous flow chamber; and,

at least a pair of generally slot-like areas in said outer shell each extending generally transversely through the inner and outer walls thereof inwardly from at least adjacent said outer shell first end toward said second end in registry with at least a portion of said inner shell whereby said heat exchanger may be mounted axially of an air flow with said air flow being heated from said first to said second temperature as it flows axially through said air flow passage and whereby said heat exchanger may be mounted transversely of an air flow with said air flow being heated from said first to said second temperature as it flows generally transversely through said air flow passage between said at least a pair of slot-like areas, said slot-like areas being generally oppositely disposed from each other intermediate said channel defining means and said outlet whereby products of combustion entering said gaseous flow chamber from said heating chamber through said channel defining means follow a serpentine path through said gaseous flow chamber to said outlet.

2. The heat exchanger as defined in claim 1 further including a thin air flow deflector surrounding said channel defining means in said air flow passage and disposed intermediate said inner and outer shells, said deflector providing means for efficiently conveying an air flow in said air flow passage axially or transversely of said heat exchanger around said channel defining means.

3. The heat exchanger as defined in claim 1 wherein said inner and outer shells have generally cylindrical cross sectional configurations, said heat exchanger further including a thin air flow deflector surrounding said channel defining means intermediate said inner and outer shells, said deflector providing means for efficiently conveying an air flow through said air flow passage either axially or transversely of said heat exchanger around said channel defining means.

4. The heat exchanger as defined in claim 3 wherein said deflector has a generally arcuate configuration over the lateral dimension thereof and is disposed in a generally parallel relationship over the surface thereof with said inner and outer shells for promoting laminar air flow around said channel defining means in air flow directions both axially and transversely of said heat exchanger.

5. The heat exchanger as defined in claim 3 wherein said deflector includes an areas disposed in intimate contact with said channel defining means to permit said deflector to further act as a heat sink.

6. In a heat exchanger of the type having a generally cylindrical inner shell defining a heating chamber and a generally cylindrical outer shell coaxially disposed relative to and spaced radially outward from said inner shell to define a generally annular air flow passage between said inner and outer shells for heating air passing therethrough and wherein said outer shell includes means for exhausting products of combustion generated in said heating chamber by a burner with said heating chamber and exhausting means being placed in gaseous flow communication with each other across said air flow passage by a cross over member, the improvement comprising:

a thin air flow deflector surrounding said cross over member in said air flow passage intermediate said inner and outer shells, said deflector having a generally arcuate configuration over the lateral dimension thereof and disposed to extend generally parallel over the surface thereof to said inner and outer shells for promoting laminar air flow around said cross over member both axially and transversely of said heat exchanger.

7. The improvement as defined in claim 6 wherein said deflector further includes an area disposed in intimate contact with said cross over member for permitting said deflector to act as a heat sink.

8. In a heat exchanger of the type employed to heat a continuous flow of air passing between the outer side wall of an inner shell defining a heating chamber and the inner side wall of an outer shell which surrounds said heating chamber wherein heat is continuously transferred at least from the outer side wall of said heating chamber to said continuous flow of air for subsequent distribution for heating enclosed areas and the like; the improvement comprising:

said inner and outer shells being generally coaxially and coextensively disposed relative to each other with said outer side wall so as to define an air flow passage therebetween over the cooperating axial lengths thereof whereby said flow of air may be selectively directed axially of said heat exchanger through said air flow passage for air heating purposes; and, said outer shell having at least a pair of slot-like areas generally oppositely disposed from each other and passing therethrough generally axially therealong from at least adjacent an outer shell first end toward an outer shell second terminal end in registry with at least a portion of said inner shell whereby said flow of air may be selectively directed transversely of said heat exchanger through said air flow passage between said slot-like areas.

9. A forced air heating system comprising in combination:

a heating system housing including at least a heated air flow outlet;

a heat exchanger recieved in said housing including an inner shell having a side wall, a closed inner end and an outer end for defining a heating chamber with an open ended outer shell generally coaxially disposed around said inner shell and radially spaced therefrom so as to define an air flow passage therebetween over the axially cooperative lengths thereof, said outer shell further including at least a pair of elongated slot areas extending therethrough from one terminal end toward the other terminal end in registry with said inner shell and means for exhausting products of combustion generated in said heating chamber, said exhausting means and heating chamber being placed in gaseous flow communication with each other across said air flow passage by a cross over member;

a fuel burner in operative communication with the outer end of said heating chamber;

means for supplying fuel to said burner to facilitate combustion within said heating chamber; and,

means for directing a flow of air to be heated from a first temperature to a second temperature elevated from the first within said system housing and through said air flow passage in one of an axial and transverse direction relative to said heat exchanger and then outwardly of said housing through said outlet wherein in said axial direction said directing means and heat exchanger are disposed such that at least a portion of said flow of air is directed from adjacent said heating chamber outermost end axially through said air flow passage and wherein in said transverse direction said directing means and heat exchanger are disposed such that at least a portion of said flow of air is directed transversely through said air flow passage between said pair of elongated slot areas in said outer shell.

10. The system as defined in claim 9 wherein said outer shell has first and second ends with said outer shell first end being generally coextensive with said inner shell closed inner end and said outer shell second end, said at least a pair of elongated slot areas comprising a pair of oppositely disposed slot areas extending from said outer shell first end toward said second end.

11. The system as defined in claim 9 further including a thin air flow deflector surrounding said cross over member in said air flow passage intermediate said inner and outer shells, said deflector providing means for efficiently conveying said air flow in said air flow passage axially or transversely of said heat exchanger around said cross over member.

12. A forced air heating system comprising in combination:

a heating system housing including at least a heated air flow outlet;

a heat exchanger received in said housing including a generally cylindrical inner shell having a side wall, a closed inner end and an outer end for defining a heating chamber with an open ended generally cylindrical outer shell generally coaxially disposed around said inner shell and radially spaced therefrom so as to define an air flow passage therebetween over the axially cooperative lengths thereof, said outer shell further including at least a pair of openings extending therethrough in registry with said inner shell and means for exhausting products of combustion generated in said heating chamber, said exhausting means and heating chamber being placed in gaseous flow communication with each other across said air flow passage by a cross over member;

a thin air flow deflector surrounding said cross over member in said air flow passage intermediate said inner and outer shells, said deflector having a generally arcuate configuration over the lateral dimension thereof and being disposed in a generally parallel relationship over the surface thereof relative to said inner and outer shells for promoting laminar air flow around said cross over member both axially and transversely of said heat exchanger;

a fuel burner in operative communication with the outer end of said heating chamber;

means for supplying fuel to said burner to facilitate combustion within said heating chamber; and,

means for directing a flow of air to be heated from a first temperature to a second temperature elevated from the first within said system housing and through said air flow passage in one of an axial and transverse direction relative to said heat exchanger and then outwardly of said housing through said outlet.

13. The system as defined in claim 12 wherein said deflector includes an area disposed in intimate contact with said cross over member to permit said deflector to further act as a heat sink.

* * * * *

United States Patent [19]

Schossow

[11] **4,241,874**

[45] **Dec. 30, 1980**

[54] **HEAT EXCHANGER**

[76] Inventor: **George W. Schossow, 2316 Lilac La., White Bear Lake, Minn. 55110**

[21] Appl. No.: **35,325**

[22] Filed: **May 2, 1979**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 798,567, May 19, 1977, Pat. No. 4,171,089.

[51] **Int. Cl.**[3] **F24B 7/04**

[52] **U.S. Cl.** ..... **237/55**; 126/112; 126/117; 165/DIG. 2; 165/104 R

[58] **Field of Search** ..... 165/140, DIG. 2, 104 R, 165/106; 237/55; 126/112, 117; 122/20 B

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Attorney, Agent, or Firm—Merchant, Gould, Smith, Edell, Welter & Schmidt

[57] **ABSTRACT**

An improved heat exchanger (74) for use with a heating furnace (72) is disclosed. The exchanger (74) includes a first plurality of tubes (84) which conduct hot exhaust gases from the furnace (72). Tubes (98) carrying outside air for combustion in the furnace (72) pass proximate the first group of tubes (84) within a heat exchange chamber (90) wherein heat is transferred to the combustion air via a heat transfer medium. Similarly, ambient air in the building passes through the heat exchange chamber (90) of the exchanger (74) through a third plurality of tubes (118). These tubes (118) also pass proximate the tubes (84) carrying hot exhaust gases. In a preferred embodiment, a conduit (80), through which the exhaust gases transit from the furnace (72) to the first plurality of tubes (84) in the heat exchanger (74), can be routed through plenums (110, 126) through which combustion air and ambient air circulate after exiting the second and third pluralities of tubes (98, 118).

18 Claims, 7 Drawing Figures

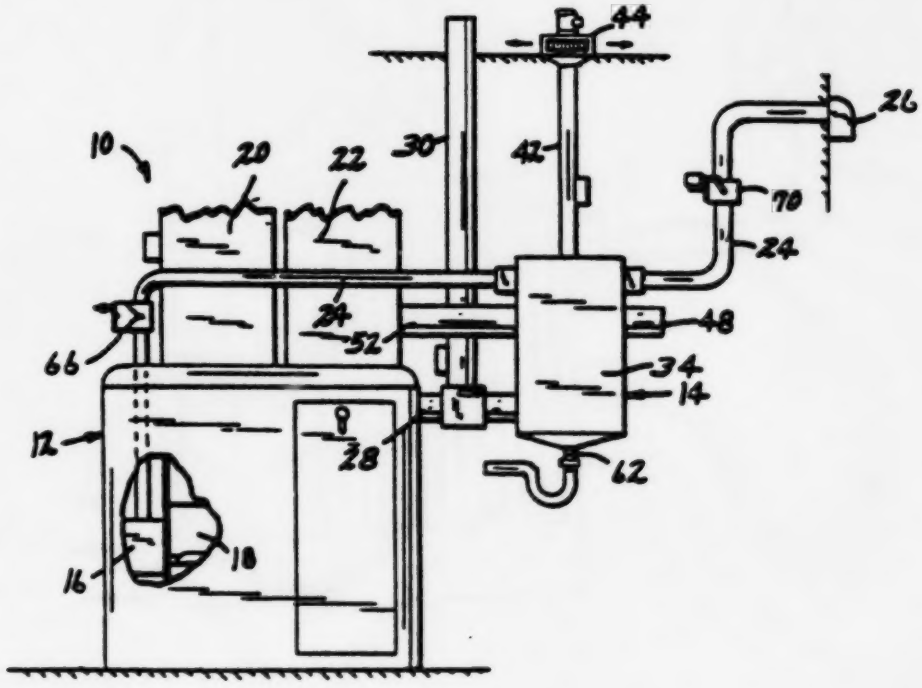

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FIG. 1

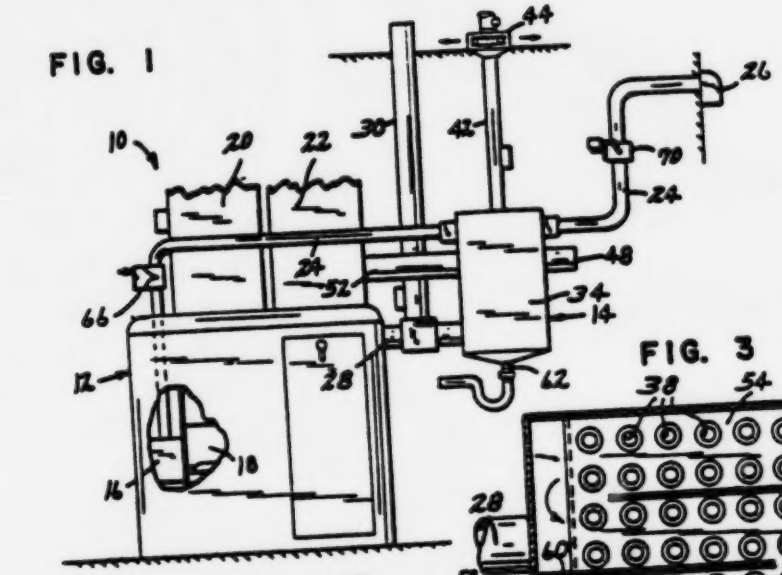

FIG. 3

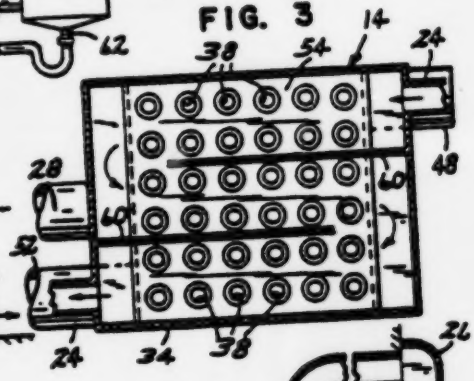

FIG. 2

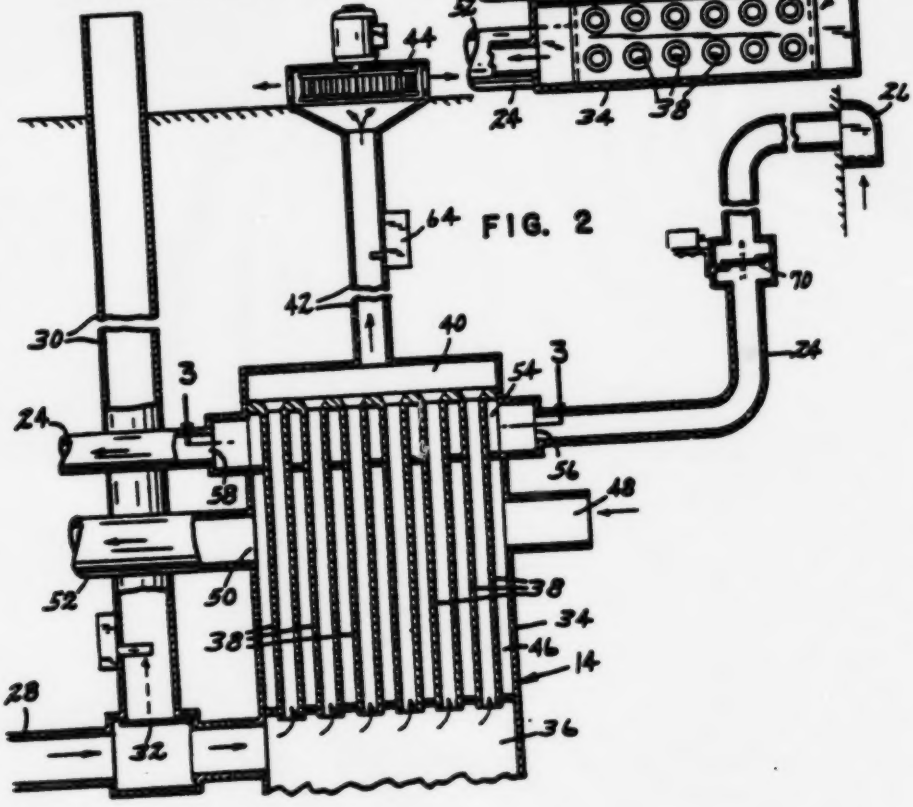

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FIG. 4

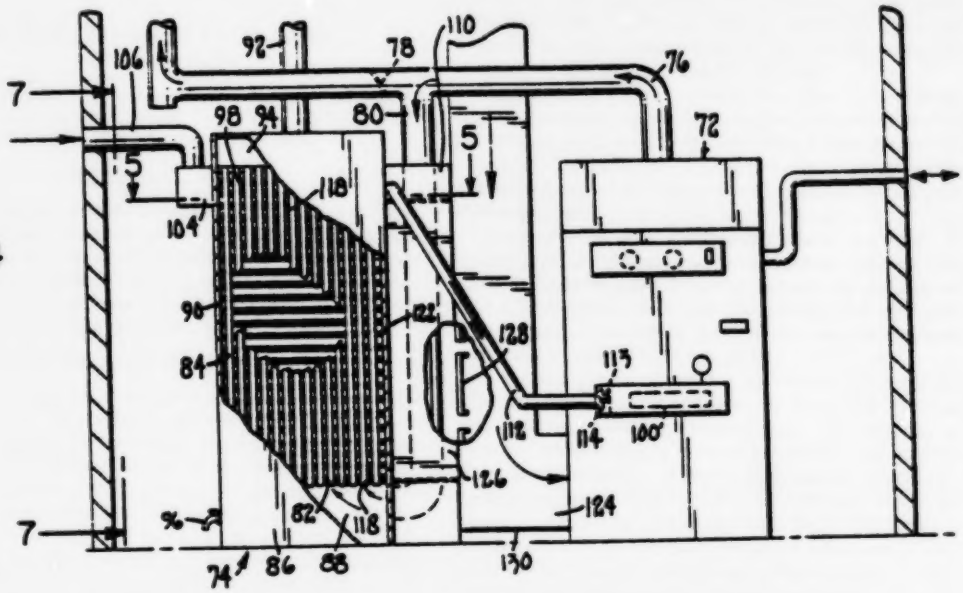
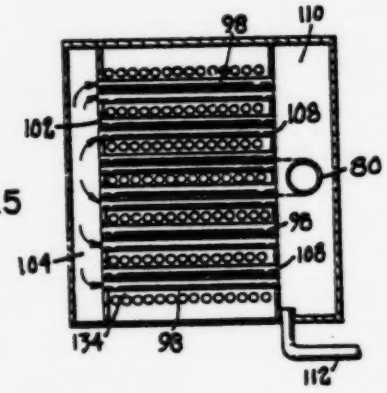


FIG. 5



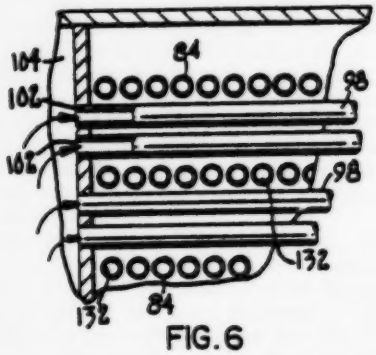


FIG. 6

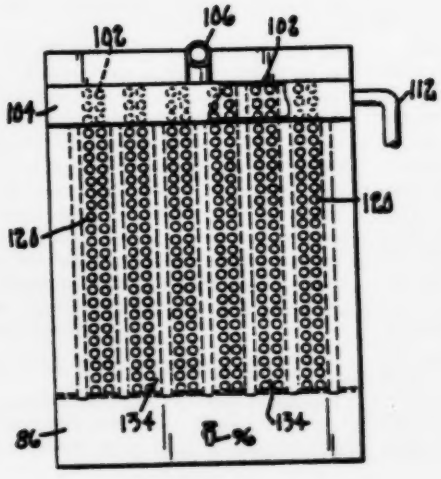


FIG. 7

HEAT EXCHANGER

CROSS-REFERENCE TO RELATED APPLICATION

This application is a continuation-in-part of applicant's copending application Ser. No. 798,567, filed May 19, 1977, now U.S. Pat. No. 4,171,089.

TECHNICAL FIELD

This invention relates generally to the field of heat exchangers used in combination with power plants, and commercial and residential heating systems. More particularly, this invention relates to the field of heat exchangers for use in home heating systems which draw cold air from outside the home for combustion.

BACKGROUND OF PRIOR ART

Home heating systems which utilize air from outside the home for combustion purposes offer significant advantages over conventional home heating systems, which draw combustion air from inside the home. Among these advantages are: (1) fuel savings due to the fact that heated room air is not being expelled up the chimney to be replaced by cold air seeping into the home; and (2) the elimination of the need for a humidifier to keep room air at a livable humidity level.

Because such systems draw cold air into their burner units rather than drawing air that has already been heated, the firebox temperature of such systems may be slightly lower than that of conventional systems. Additionally, as the stack gases proceed up the chimney, moisture contained in those gases condenses on the inside of the chimney, where it may cause chimney freeze up. Finally, the hot combustion gases being exhausted up the chimney represent wasted heat, which could be used for further room heating.

It is known in conventional furnace systems to utilize the hot stack gases for room heating by diverting the stack gases through a heat exchanger which distributes some of the heat from the flue gases to inside air. One such system is disclosed in U.S. Pat. No. 3,813,039. Heretofore, however, there has not been a furnace system which not only utilized the heat from flue exhaust gases to heat up air circulating within the home, but also was adapted to draw cold air from outside the home for combustion purposes, raise the temperature of that air slightly so as to increase the temperature of the firebox, and decrease the temperature and moisture content of the flue gases.

In the past, industrial combustion systems, such as coal-burning power plants, have utilized cyclone separators, electrostatic precipitators, and wet scrubbers to clean the stack gases of fly ash and pollutants. One of the problem pollutants has been sulphur, which is exhausted to atmosphere as sulphur dioxide. Power plants and commercial boilers thus utilize expensive, extremely tall chimney stacks to exhaust noxious gases well away from residential levels. Until the present invention, a system has not been known which satisfactorily scrubs the pollutants from the stack gases, eliminating the need for very high chimneys, and retrieves heat energy for subsequent work.

BRIEF SUMMARY OF THE INVENTION

The present invention is an improved heat exchanger which can be used in combination with furnace systems. It is particularly adaptable to be used in conjunction with home furnaces. Exhaust gases from the furnace are cooled as they pass through the exchanger, and moisture and other impurities can be removed from the exhaust by the exchanger prior to venting from the building in which the furnace is housed.

Additionally, outside air which is used for combustion in the burner of the furnace can be warmed by the exchanger prior to its being introduced into the burner chamber. Similarly, ambient air in the vicinity of the exchanger can be pre-warmed by passing it through the exchanger prior to recirculating it through the furnace.

The improved heat exchanger includes a cabinet or housing. A first plurality of tubes extends through this cabinet. Exhaust gases from the furnace are introduced into a first end of each tube comprising this plurality. The exhaust is, thereafter, permitted to pass through the tubes and is vented to the outside.

A second group of tubes also extends through the cabinet. Outside air is admitted to first ends of these tubes and exits from the second ends. This outside air then is conveyed to the furnace burner for combustion. Each of a third group of tubes which passes through the cabinet is open at its first end to ambient air within the space in which the exchanger is located. This ambient air passes through the third group of tubes and is then joined into the air circulation system of the furnace prior to being reheated and recirculated throughout the building in which the furnace is located.

The cabinet is filled with a heat transfer medium, and that medium surrounds portions of all of the tubes which extend within the cabinet. The medium serves to conduct heat from the hot exhaust gases passing through the first plurality of tubes to the combustion air and recirculation air which pass through the second and third groups of tubes respectively. In the preferred embodiment, the heat transfer medium is a liquid, and it has been found that ordinary tap water can effectively function as the medium.

In the preferred embodiment, the exhaust gases from the furnace can be introduced into the first plurality of tubes by way of a conduit interconnecting the furnace exhaust system and said first tubes. Existence of this conduit affords an opportunity to further heat the outside air for combustion and the ambient air. After exiting the tubes by which they pass through the heat exchanger, they can be made to pass through first and second outlet plenums respectively. A second stage of heating can be accomplished if the exhaust conduit is run through these plenums.

The heat exchanger of the present invention increases both the efficiency of a furnace with which it is used and also increases the efficiency of the entire heating system. In most home heating furnaces, a significant portion of the heat generated by the furnace is lost as exhaust gas exiting up a chimney. The use of the heat exchanger recovers most of this heat so that the exhaust gases exit up the chimney at a relatively low temperature. As will be illustrated in the detailed description, the heat exchanger returns the extracted heat back into the home and the furnace.

Normal flue gases contain a certain amount of steam. The heat exchanger of the present invention condenses the steam vapor out as a liquid. Thermal energy is thus gained by the heat exchanger by this process.

In conventional heating systems an after draft exists after the fuel burning process has been stopped. This after draft exists because the chimney has been heated

by hot exhausts during the burning process. By utilization of the heat exchanger of the present invention, the chimney is kept cool and, hence, the after draft is reduced. A further gain in the efficiency of the system is thus attained.

In some heating systems, a barometric regulator is provided to cool a chimney should the chimney become too hot. The barometric regulator permits house air to enter the chimney to cool it. Again, by utilizing the heat exchanger of the present invention, the chimney is kept cool and, hence, the barometric regulator would not operate or would not be needed. A further gain in efficiency is thus attained.

The invention of this application thus is an improved heat exchanger for use with heating furnaces. The specific advantages of the invention will become apparent with reference to the accompanying drawings, detailed description of the invention, and claims.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a view in side elevation, with portions thereof broken away, showing a furnace/heat exchanger combination;

FIG. 2 is an enlarged view in section of a portion of FIG. 1, with portions thereof broken away;

FIG. 3 is a view in section taken along the line 3—3 of FIG. 2, and indicating the flow of combustion air through the upper chamber of the heat exchanger;

FIG. 4 is a view in side elevation, with portions broken away, showing a heat exchanger in accordance with the present invention in combination with a heating furnace;

FIG. 5 is a view in section taken along the line 5—5 of FIG. 4, and indicating combustion air flow;

FIG. 6 is an enlarged view of a portion of FIG. 5, some portions broken away;

FIG. 7 is a front view of the present invention with portions thereof broken away.

DETAILED DESCRIPTION OF THE INVENTION

Referring first to FIG. 1, there is disclosed a furnace/heat exchanger combination, generally designated by the numeral 10, in which a furnace 12 is coupled with a heat exchanger 14 constructed in accordance with the present invention. Furance 12 comprises a burner unit 16, including a blower or fan (not shown) which draws air to be mixed with the fuel for combustion. A flame burns in firebox 18 to heat air which is subsequently distributed through the home via duct 20, and which is returned to the furnace for reheating via duct or plenum 22.

Burner unit 16 draws combustion air from outside the home via conduit 24, which extends from burner unit 16 to vent 26, which is located outside the home. The gaseous products of combustion are exhausted from furnace 12 via exhaust conduit 28, and may be selectively exhausted up chimney stack 30 or routed through heat exchanger 14. A bypass gate 32 is used to select which route exhaust gases will take.

Exhaust gases enter heat exchanger 14 when gate 32 closes chimney 30, as clearly shown in FIG. 2. Heat exchanger 14 comprises an outer wall 34 which encloses four chambers. The lowermost chamber 36 is a plenum into which the gaseous products of combustion flow via exhaust conduit 28. A plurality of vertically oriented tubes 38 are in fluid communication with lower chamber 36. Heat exchanger tubes 38 are of a type known in the art, such as those constructed with a copper outer wall and a corrosion-resistant inner wall of ceramic or Teflon. As indicated in FIG. 2, the exhaust gases flow upwardly through tubes 38. At their upper most ends, tubes 38 are in fluid communication with chimney stack 42. Mounted in the mouth of chimney stack 42 is a driven fan 44. The action of heat exchanger 14 causes the flue gases to cool markedly before reaching chimney 42, thus reducing the volume of the flue gases by 50% to 70%, with the result that fan 44 need not be a large power consuming fan in order to create the desired negative pressure in chimney 42. Since fan 44 is mounted downstream of heat exchanger 14, a negative pressure is created within tubes 38, which induces a draft upwardly through the heat exchanger so as to draw the exhaust gases through exhaust conduit 28, into lower chamber 36, through tubes 38 into chamber 40, and upwardly through chimney 42, and thus out of the house.

There are two intermediate chambers in heat exchanger 14. The lowermost of these chambers 46 has an upstream inlet 48, which is open to room air. Chamber 46 has a downstream outlet 50 which is in communication with the return air plenum 22 of the furnace via duct 52. The return air blower of the furnace (not shown) thus draws air from inside the home in the vicinity of the heat exchanger through chamber 46, where it circulates around tubes 38 before passing into the recirculation system of the furnace.

The uppermost intermediate chamber, indicated by the numeral 54, has upstream inlet 56 and downstream outlet 58 into which combustion air conduit 24 enters and exits.

Both chambers 46 and 54 are partitioned by vertically extending baffles 60, as shown in FIG. 3, so as to insure complete circulation around tubes 38 of the air entering and exiting those chambers.

Heat exchanger 14 includes a drain 62 located at its lower end to drain away the moisture which condenses and runs down tubes 38 into chamber 36.

Mounted in chimney stack 42 is a pressure sensitive switch 64 which is operably connected to burner unit 16 by means not shown, but well known in the art, and is adapted to prevent burner unit 16 from igniting, and to turn burner 16 off, if the appropriate negative pressure does not exist within chimney stack 42, i.e. if a draft up through the heat exchanger does not exist. A barometric draft regulator 66 can be mounted in a tee section of conduit 24 shown in FIG. 1, and a butterfly valve 70 which is normally open, but closes after a time delay following shut off of burner unit 16, can be included.

In operation, thermostatic control means (not shown) turn on fan 44 and open butterfly valve 70. When the appropriate draft up chimney 42 is sensed by switch 64, burner unit 16 is ignited. Cold air from outside the home is drawn into burner unit 16 via conduit 24 for combustion purposes. This cold air circulates through chamber 54 of heat exchanger 14 before passing into burner unit 16. The gaseous products of combustion are exhausted from firebox 18 via conduit 28 into lower chamber 36 of heat exchanger 14, where they are drawn up through tubes 38 and exit via chimney stack 42. The return air blower (not shown) of furnace 12 draws air from inside the home in through chamber 46 of heat exchanger 14 before entering the recirculation system of furnace 12. As the hot stack gases pass upwardly through tubes 38, cool inside air circulates around their lower portions where it picks up heat from them. At this point, the

slightly acidic moisture carried within the exhaust gases condenses on the walls of tubes 38 and is drawn by gravity downwardly into chamber 36 and carried away via drain 62. This action of the moisture keeps tubes 38 clean. The cold combustion air traveling through conduit 24 is heated as it passes through chamber 54 and thus passes into burner unit 16 in a warmed state.

Referring now to FIGS. 4–7, a furnace 72 is shown in combination with a second embodiment of the heat exchanger 74 of the present invention. The furnace may be of the same type as previously discussed herein.

Gaseous products of combustion in the furnace 72 are exhausted from the furnace 72 by way of an exhaust conduit 76. These combustion products can either be vented outside the building directly or channeled through the heat exchanger 74. Channeling of the gases is selectively controlled by a bypass gate 78. If the gases are directed through the heat exchanger 74, the bypass gate 78 is closed so that they pass through a branch portion 80 of the exhaust conduit 76. This branch portion 80 of the exhaust conduit 76 introduces the gases from the furnace 72 at the first ends 82 of a plurality of tubes 84 mounted within a cabinet or housing 86 of the exchanger 74. Introduction may be made by allowing the exhaust products in the branch conduit 80 to fill a first chamber 88 which encloses the entrances to the first ends 82 of the plurality of tubes 84. The exhaust is, thereafter, drawn through the tubes 84, which extend through an intermediate heat exchange chamber 90, and exhausted outside the building in which the heat exchanger 74 is housed via an exchanger venting conduit 92. The draft in the tubes 84 may be created by a fan (not shown) and a chimney extension of the venting conduit 92, as shown at 42 in FIG. 1, can be provided by which actual venting occurs. Confluence of the gases after exiting individual tubes 84 may occur in a second chamber 94 which can be placed in fluid communication with the venting conduit 92.

Although not essential to the invention, the best mode contemplates that the tubes 84 be straight and oriented generally vertically. Such orientation would position the tubes 84 parallel with respect to one another. In such a configuration, the first chamber 88 and first ends 82 of the tubes 84 would be positioned at the lower end in order that the natural tendency of the hot gases to rise would facilitate venting.

Means can be incorporated for draining moisture of condensation, which may form in a manner hereinafter described, from the tubes 84. The means may comprise a spiggot 96 mounted externally on the first chamber 88 for controlling fluid flow from inside the chamber 88.

The exhaust gases will thus provide a source of heat to be transferred within the exchanger 74. Both oxygen-rich cool air for combustion drawn from external to the building in which the exchanger 74 is housed and ambient air inside the building will receive this heat provided by the exhaust gases. A second plurality of tubes 98 extends through the intermediate chamber 90 positioned between the first and second chambers 88, 94 heretofore described. This second plurality 98 is provided to channel outside air through the exchanger 74 prior to its being conveyed to the furnace burner 100. First ends 102 of this second plurality of tubes 98 are in fluid communication with outside air as through an inlet plenum 104 and means for conducting outside air into said inlet plenum 104, such as combustion air intake 106 in FIG. 4.

The combustion air is then drawn through the second plurality of tubes 98 and thence conveyed to the furnace burner 100 for combustion. Enclosing second ends 108 of this second plurality of tubes 98 may be a first outlet plenum 110 into which the combustion air exits from the tubes 98. Piping 112 connects the first outlet plenum 110 to an air inlet box 113. A burner fan 114 may be provided in the air inlet box 113 in order to insure a flow of outside combustion air to the burner 110.

A third plurality of tubes 118 pass through the intermediate chamber 90. First ends 120 of the tubes 118 open to ambient air inside the room. Ambient air passed through these tubes 118 is transmitted to the return air blower of the furnace system. By passing this ambient air through the heat exchanger 74, air having been circulated throughout the building in which the furnace is located and having become cool would be preheated by the air passing through the tubes 118 prior to being reheated by the fire box (not shown).

The ambient air can be transmitted from the second ends 112 of the third plurality of tubes 118 to the air recirculation system 124 via a second outlet plenum 126. An apertured wall 128 between the second outlet plenum 126 and the return recirculation duct 130 insures flow of ambient air through the third plurality of tubes 118. Essentially, the return recirculation duct 130 would be in fluid communication with the ambient air about the heat exchanger 74 through the third plurality of tubes 118. Forced flow through the duct 130 will have an eduction effect as it passes the apertured wall 128 and will draw the ambient air therethrough.

A heat exchange medium is provided in the intermediate chamber 90 surrounding portions of the first, second and third pluralities of tubes 84, 98, 118. Heat will be transferred through this medium from the hot exhaust gases in the first plurality of tubes 84 to the air in the second and third pluralities 98, 118. Cooling of the exhaust may cause condensation within the first plurality of tubes 84. If the first plurality of tubes 84 is oriented other than horizontally, this condensation will be made to flow down the inner walls 132 of the tubes 84 and will have a scrubbing effect thereon as previously described. The condensation will run down into the first chamber 88 and can be drained therefrom.

It will be clear to one of skill in the art that one of any number of fluids can be used as a heat conductive medium. It has been found that ordinary tap water, which is inexpensive and readily available, adequately serves this function.

In order to maximize the transfer of heat, the first plurality of tubes 84 can be arranged in parallel spaced rows. So spacing the rows will define at least one corridor 134 therebetween. The tubes comprising the second and third pluralities 98, 118 can be made to extend through these corridors 134 in order to maximize efficiency. Heat transfer can be further improved if the tubes comprising the second and third pluralities 98, 118 are oriented generally transverse to the orientation of the first plurality 84.

Warming of combustion air and ambient room air can be augmented by a second warming stage. The branch conduit 80 by which the hot exhaust gases are introduced into first ends 82 of the first plurality of tubes 84 can be extended through the first and second outlet plenums 110, 126. Passage of combustion air and ambient air about the conduit 80 in the first outlet plenum 110 and the second outlet plenum 126 respectively will cause those gases to take on even more heat.

Numerous characteristics and advantages of my invention have been set forth in the foregoing description. It will be understood, of course, that this disclosure is, in many respects, only illustrative, and changes may be made in details, particularly in matters of shape, size and arrangement of parts. The scope of the invention is defined in the language in which the appended claims are expressed.

What is claimed is:

1. A heat exchanger for use with a heating furnace, comprising:

a cabinet;

a first plurality of tubes extending through said cabinet, each of said tubes having first and second ends;

means for introducing exhaust gases from the furnace into first ends of said first plurality of tubes;

a second plurality of tubes extending through said cabinet, each of said second plurality having first and second ends, said first end of said second plurality being in fluid communication with outside air;

means for conveying outside air in said second plurality of tubes from said second ends to the furnace burner;

a third plurality of tubes extending through said cabinet, said third plurality having first ends in fluid communication with ambient air inside the home;

means for transmitting air inside said third plurality of tubes from second ends thereof to the return air blower of the furnace; and

heat transfer medium in said cabinet surrounding portions of said first, second, and third pluralities of tubes disposed within said cabinet.

2. The heat exchanger of claim 1 wherein said conveying means includes a first outlet plenum and said transmitting means includes a second outlet plenum.

3. The heat exchanger of claim 2 wherein said means for introducing exhaust gases from the furnace into said first plurality of tubes comprises a first chamber in fluid communication with first ends of said first plurality of tubes and a conduit interconnecting said chamber and the furnace.

4. The heat exchanger of claim 3 wherein said conduit passes through said first and second outlet plenums.

5. The heat exchanger of claim 1 wherein one of said first and second ends of said first plurality of tubes is elevated with respect to said other end.

6. The heat exchanger of claim 5 further comprising means proximate said lower end of said first plurality of tubes for draining moisture of condensation therefrom.

7. The heat exchanger of claim 1 further comprising a chimney in fluid communication with said second ends of said first plurality of tubes.

8. In combination with a furnace system wherein the furnace includes a burner which draws combustion air from outside the building in which the furnace is housed, a heat exchanger for cooling exhaust gases from the furnace and removing moisture therefrom before the exhaust gases are vented from the building, for warming outside combustion air before it reaches the burner, and for pre-warming cool inside air before recirculating it through the furnace, comprising:

a conduit carrying exhaust gases from the furnace;

a housing including first and second chambers and an intermediate chamber positioned between said first and second chambers, said first chamber in fluid communication with said conduit;

a first plurality of tubes having first and second ends in fluid communication with said first and second chambers respectively, said first plurality extending through said intermediate chamber;

a first outlet plenum enclosing a first portion of said conduit;

a second outlet plenum enclosing a second portion of said conduit;

an inlet plenum;

means for conducting outside air for combustion into said inlet plenum;

a second plurality of tubes extending through said intermediate chamber, said tubes having first ends in fluid communication with said inlet plenum and second ends in fluid communication with said first outlet plenum;

a third plurality of tubes extending through said intermediate chamber, said third plurality having first ends in fluid communication with ambient air inside the building in which the furnace is located and second ends in fluid communication with said second outlet plenum;

liquid medium disposed in said intermediate chamber, said medium surrounding portion of said first, second, and third pluralities of tubes extending through said intermediate chamber;

means for conveying combustion air in said first outlet plenum to the furnace burner;

means for transmitting ambient air in said second outlet plenum to the return air blower of the furnace; and

means for draining moisture of condensation from said first plurality of tubes.

9. The heat exchanger of claim 8 wherein said inlet plenum and said first outlet plenum are mounted on said housing adjacent said intermediate chamber and enclose first and second ends respectively of said second plurality of tubes.

10. The heat exchanger of claim 9 wherein said second outlet plenum is mounted on said housing adjacent said intermediate chamber and encloses second ends of said third plurality of tubes.

11. The heat exchanger of claim 10 wherein said second outlet plenum is disposed beneath said first outlet plenum.

12. The heat exchanger of claim 8 wherein each of said first plurality of tubes is straight and is oriented generally vertically and parallel with respect to each other of said plurality.

13. The heat exchanger of claim 12 wherein said first plurality of tubes is arranged in parallel spaced rows.

14. The heat exchanger of claim 13 wherein spacing of said parallel rows of said first plurality of tubes defines at least one corridor extending through said intermediate chamber, and wherein said second and third pluralities of tubes extend through said corridor.

15. The heat exchanger of claim 14 wherein said tubes comprising said second and third pluralities are disposed generally transverse to said tubes comprising said first plurality.

16. The heat exchanger of claim 13 wherein said first ends of said first plurality of tubes are disposed beneath said second ends.

17. The heat exchanger of claim 16 wherein said means for draining moisture of condensation is disposed within said first chamber.

18. The heat exchanger of claim 8 wherein said liquid medium is water.

* * * * *

APPENDIX C

HOT WATER TO AIR

FINNED TUBE HEAT EXCHANGER

This appendix presents calculations that were carried out in order to size the finned tube heat exchanger referred to in Chapter 5. The operating inlet and outlet design temperatures that were used to size this heat exchanger are presented in Figure C.1. Energy is transferred in this unit at the rate of 95% of the burner input rate (22.3 kW or 76,000 BTU/hr).

The heat exchanger sizing was carried out according to the methods presented in the AEROFIN WATER COILS FOR HEATING bulletin MP-72. These calculations are presented using the same nomenclature and system of units presented in that bulletin. Portions of bulletin MP-72 are included at the end of this appendix.

The AEROFIN bulletin specifies that the temperature of the water leaving the heat exchanger must be at least 15F above the temperature of the air leaving the heat exchanger in order that their design calculations will be valid. This is not the case with the total heat exchanger as shown in Figure C.1. In order to meet this prerequisite, the total heat exchanger was analysed as a series of identical heat exchangers, each with the same effectiveness, arranged in an overall counterflow air-water flow pattern. Each of these heat exchangers can be viewed as a bank of finned tubes in a multi-bank heat exchanger.

In order to achieve the required 15F temperature difference it was necessary to consider four heat exchanger sections. The design temperatures of all entrance and exit flows are shown in Figure C.2.

The rate of heat transfer can be specified as a function of water inlet temperature, $T_{w,i}$, water outlet temperature, $T_{w,o}$, and the mass flow rate of water as shown in equation C.1.

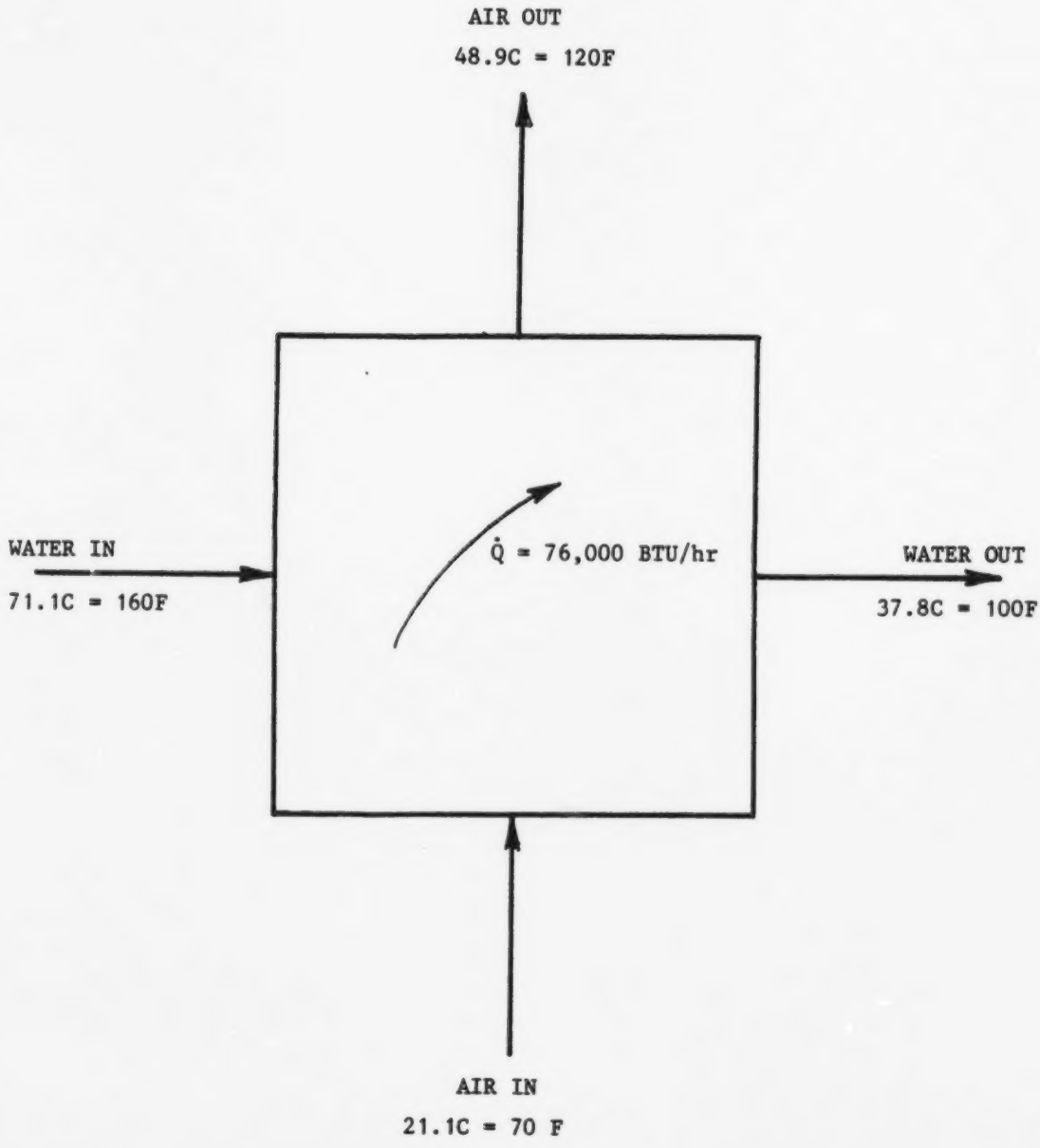


Figure C.1: Finned Tube Heat Exchanger Design Temperatures

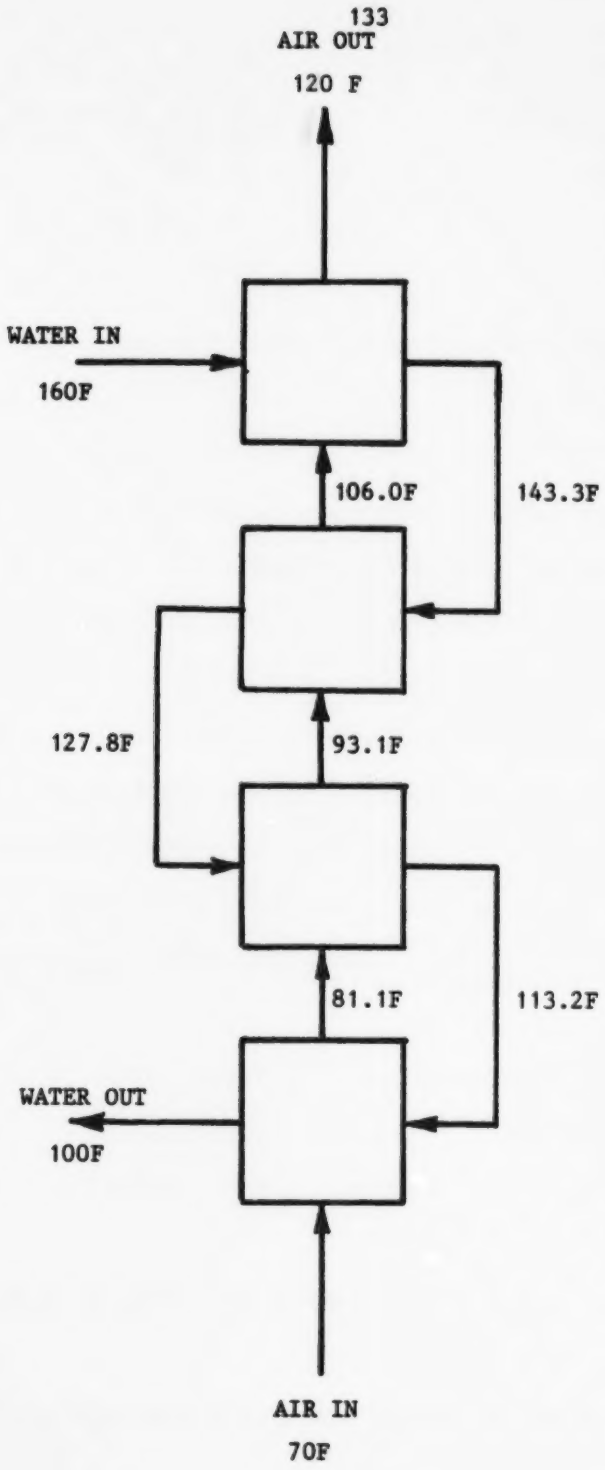


Figure C.2: Design Temperatures of Finned Tube Heat Exchanger Tube Banks

$$\dot{Q} = \dot{m}_w C_{p,w}(T_{w,i} - T_{w,o}) \qquad (C.1)$$

where $C_{p,w}$ = 1 BTU/lbm F

Solving equation C.1 ($\dot{Q}$ = 76,000 BTU/hr) yields a water flow rate of 2.11 GPM or 2.53 USGPM.

Similarly, solving for the air flow rate yields a flow of 6333 lbm/hr or 1487 CFM (@ air density of 0.071 lbm/ft³). Assuming an air flow face area of 3 ft² gives an air velocity of 8.26 ft/sec or 495 ft/min.

Since each of the tube bank sections is identical to the other three, it was necessary only to size one section. Consider the section where water enters at 160F and air enters at 106F.

$$F_w = \frac{\text{air temperature rise}}{\text{average water temperature - average air temperature}} = 0.36$$

From curve #2, Pg. 11 of bulletin MP-72 (one row coil):

minumum water velocity = v_w = 2.4 ft/sec

maximum number of circuits = USGPM x 1.21 / v_w = 1.28

Therefore, each tube bank should have one circuit for water flow.

Referring to Table 6, Pg. 10 of bulletin MP-72, the required tube bank size is specified by a nominal tube length of 30 inches and a ten tube face.

Thus, the total heat exchanger would consist of four of these heat exchangers connected in series and in counterflow as shown in Figure C.2. The total heat exchanger could also be constructed of a pair of two row heat exchangers which are also available from AEROFIN.

Table 5 (Pg. 10) of the bulletin MP-72 indicates a pressure drop on the air side of the heat exchanger of 0.083 inches H_2O per row indicating a total of 0.33 inches H_2O for the four rows of heat exchanger.

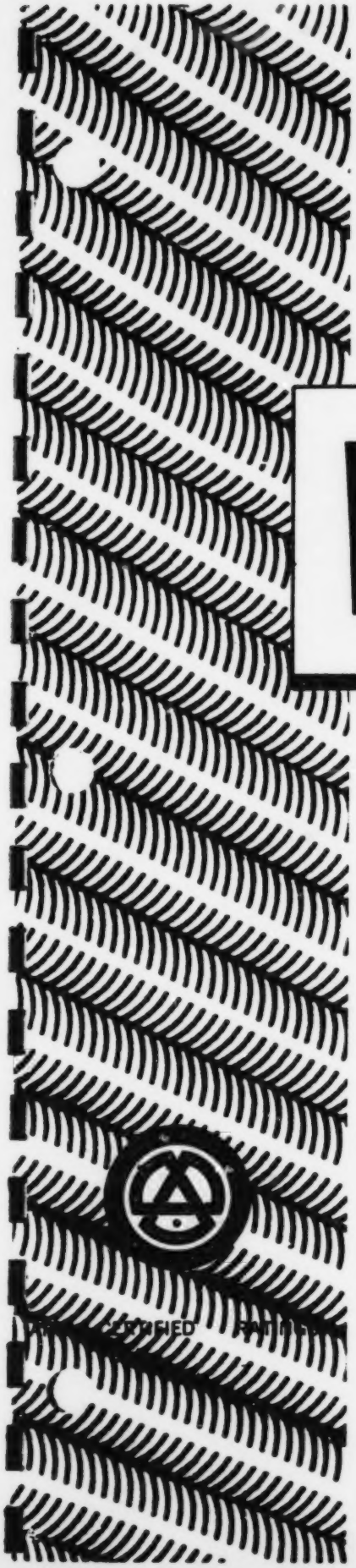
Using Table 7 (Pg. 10) and Curve 4 (Pg. 13) of the bulletin MP-72, the pressure drop of water passing through one row (or bank) of tubes is 1.045 x 3.6 ft H_2O or about 3.76 ft H_2O. Since there are four tube banks connected in series, the total water side pressure drop is about 15 ft H_2O. The power expended in pumping this water can be quantified as the pressure drop multiplied by the volumetric flow rate.

$$\text{pumping power} = \dot{V}\,\Delta P \cong \frac{1}{100} \text{ horsepower} = 7.5 \text{ watts}$$

AEROFIN

WATER COILS

For Heating



AEROFIN CORPORATION

4621 MURRAY PLACE LYNCHBURG, VIRGINIA

AEROFIN CORPORATION (Canada) LTD.

Gananoque, Ontario

TYPE MP

All dimensions approximate. Certified drawings upon request.

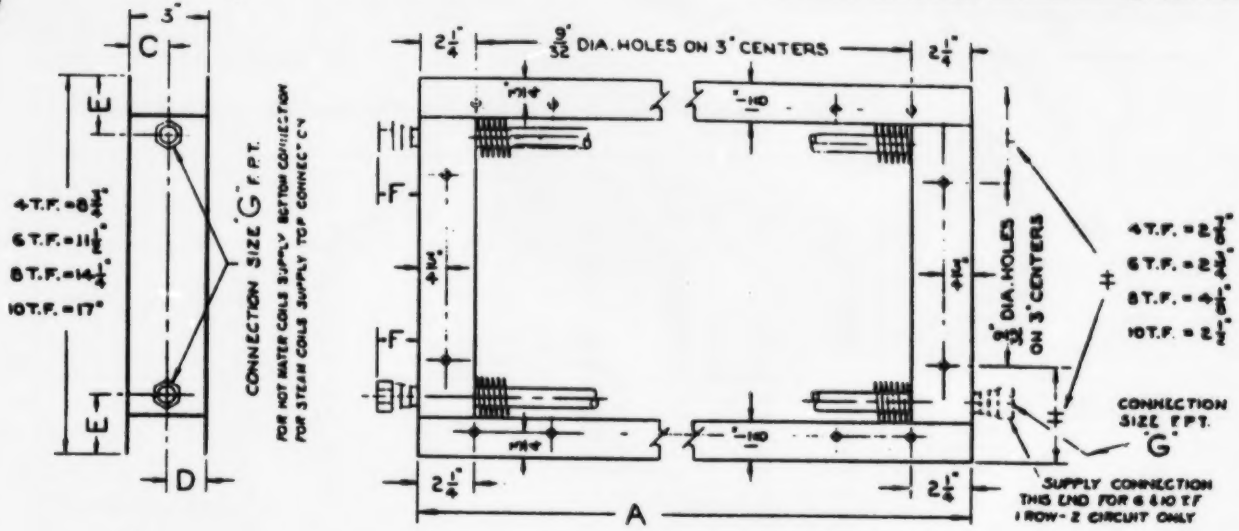


TABLE 1 — One and Two Rows — 4, 6, 8 and 10 Tube Face

Nominal Tube Length	1'-0"	1'-3"	1'-6"	1'-9"	2'-0"	2'-3"	2'-6"
A	1'-4 1/2"	1'-7 1/2"	1'-10 1/2"	2'-1 1/2"	2'-4 1/2"	2'-7 1/2"	2'-10 1/2"

TABLE 2 — One and Two Rows — 4, 6, 8 and 10 Tube Face

ONE ROW									
One Circuit					Two Circuit				
C	D	E	F	G	C	D	E	F	G
1 1/2"	1 1/2"	2 1/4"	1/2"	1/2"	1 1/2"	1 1/2"	2 1/4"	1/2"	1/2"

TWO ROW									
One Circuit					Two Circuit				
C	D	E	F	G	C	D	E	F	G
1 1/2"	1 1/2"	1 1/2"	1/2"	1/2"	1"	1"	2"	1/2"	1/2"

TABLE 3

NUMBER OF CIRCUITS

TUBE FACE	ROWS DEEP	
	ONE	TWO
4	1 OR 2	1 OR 2
6	1 OR 2*	1 OR 2
8	1 OR 2*	1 OR 2
10	1 OR 2*	1 OR 2

* OPPOSITE END CONNECTIONS

TABLE No. 4 — Face Area (Square Feet)

Overall Casing Height	Tube Face	NOMINAL TUBE LENGTH						
		1'0"	1'3"	1'6"	1'9"	2'0"	2'3"	2'6"
8 3/4"	4	.48	.60	.72	.84	.96	1.07	1.19
11 1/2"	6	.71	.89	1.06	1.24	1.42	1.60	1.77
14 1/4"	8	.94	1.17	1.41	1.65	1.88	2.12	2.35
17 "	10	1.17	1.46	1.76	2.05	2.44	2.64	2.93

TABLE No. 5 — Air Friction (Inches of Water)

Rows Deep	AIR FACE VELOCITY—FEET PER MINUTE†									
	300	400	500	600	700	800	900	1000	1100	1200
1	.036	.058	.083	.113	.145	.183	.22	.265	.31	.36
2	.074	.12	.17	.23	.30	.37	.45	.54	.63	.72

†Measured at 70°F & 29.92" Barometer.

TABLE No. 6 — Volume of Air (CFM) To Be Passed Through Coil

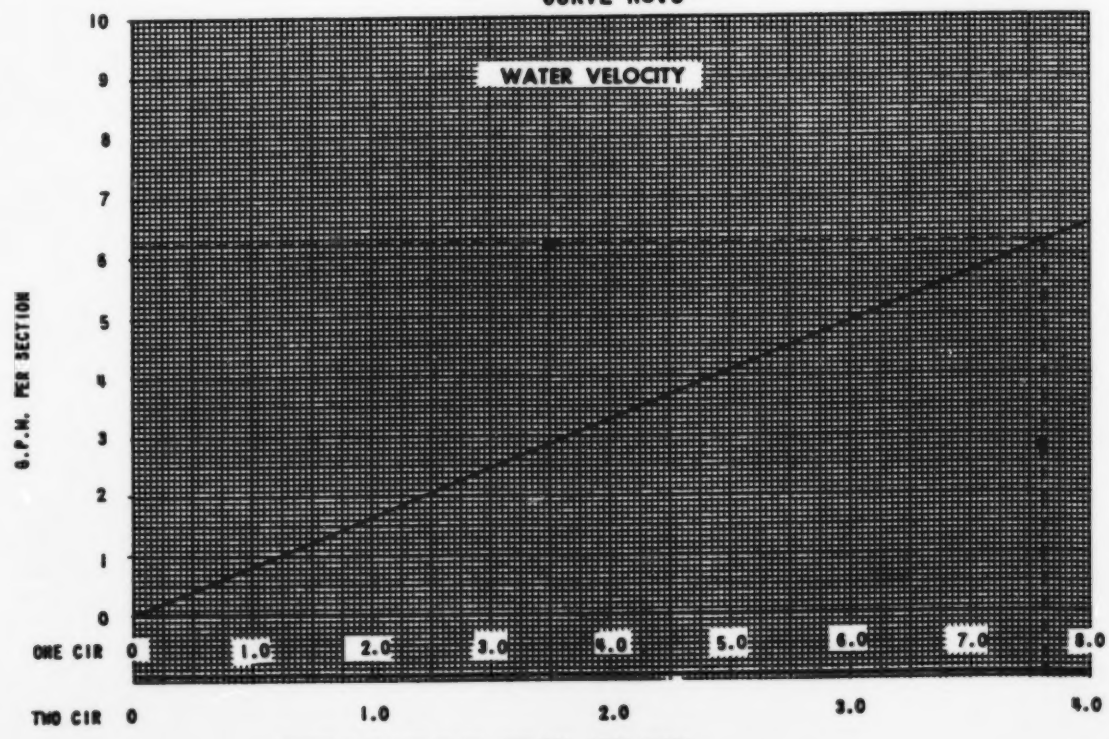
Overall Casing Width	Tube Face	Air Face Velocity	NOMINAL TUBE LENGTH						
			1'0"	1'3"	1'6"	1'9"	2'0"	2'3"	2'6"
8¾"	4	300	130	180	210	240	300	330	360
		400	200	240	280	320	400	440	480
		500	250	300	350	400	500	550	600
		600	300	360	420	480	600	660	720
		700	350	420	490	560	700	770	840
		800	400	480	560	640	800	880	960
11½"	6	300	210	360	330	360	420	480	540
		400	280	280	440	480	560	640	720
		500	350	450	550	600	700	800	900
		600	420	540	660	720	840	960	1080
		700	490	630	770	840	980	1120	1260
		800	560	720	880	960	1120	1280	1440
14¼"	8	300	300	360	420	480	570	630	690
		400	400	480	560	640	760	840	920
		500	500	600	700	800	950	1050	1150
		600	600	720	940	960	1140	1260	1380
		700	700	840	980	1120	1330	1470	1610
		800	800	960	1120	1280	1520	1680	1840
17 "	10	300	360	450	540	600	690	780	870
		400	480	600	720	800	920	1040	1160
		500	600	750	900	1000	1150	1300	1450
		600	720	900	1080	1200	1380	1560	1740
		700	840	1050	1260	1400	1610	1820	2030
		800	960	1200	1440	1600	1840	2080	2320

TABLE No. 7

Pressure Drop Correction Factor F

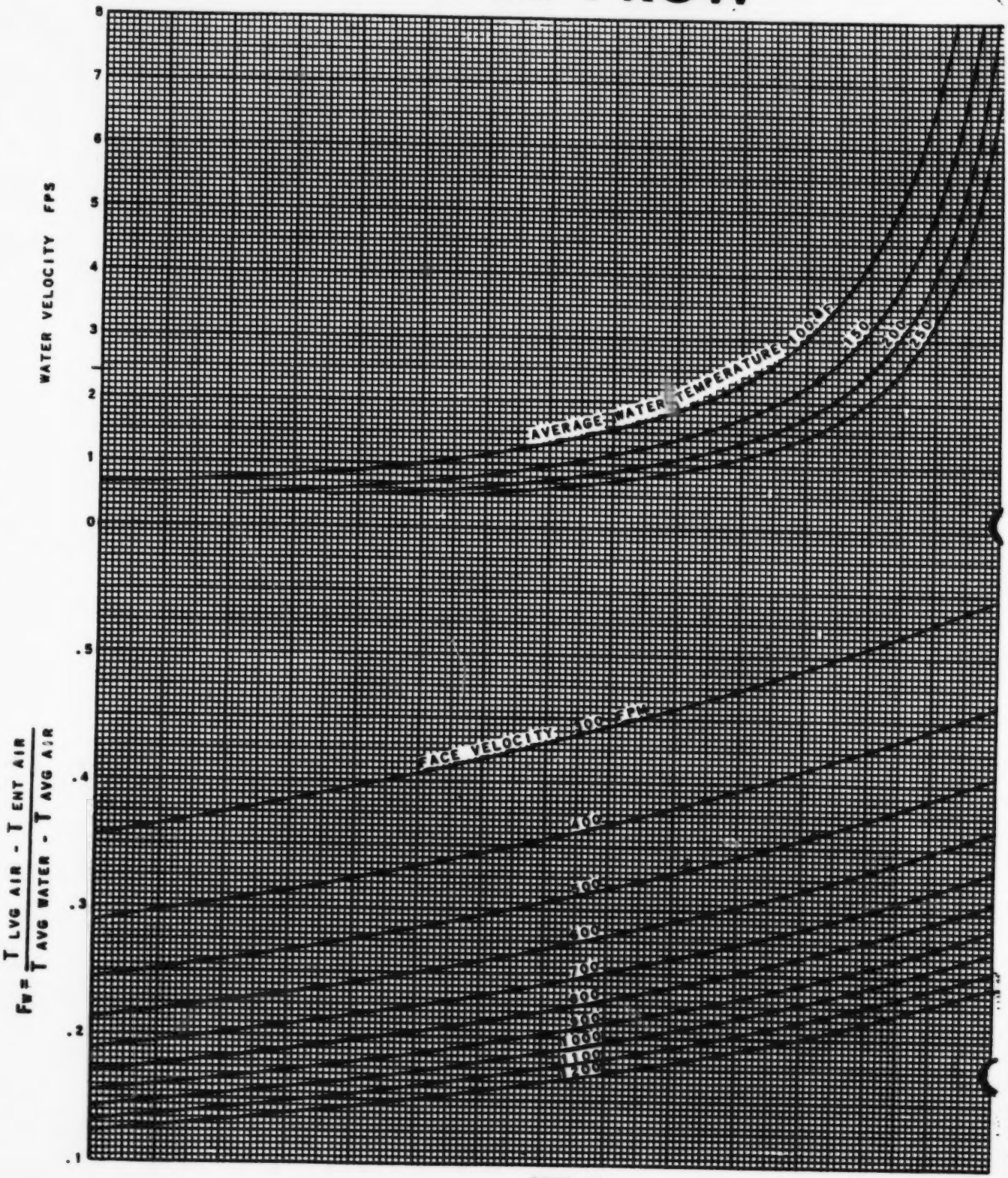
Average Temp.	Factor	Average Temp.	Factor
90	1.152	180	.935
100	1.113	190	.920
110	1.088	200	.900
120	1.070	210	.885
130	1.045	220	.870
140	1.020	230	.855
150	1.000	240	.835
160	.975	250	.823
170	.955		

CURVE No.5



CURVE No. 2 139

PERFORMANCE RATINGS
TYPE MP 1 ROW

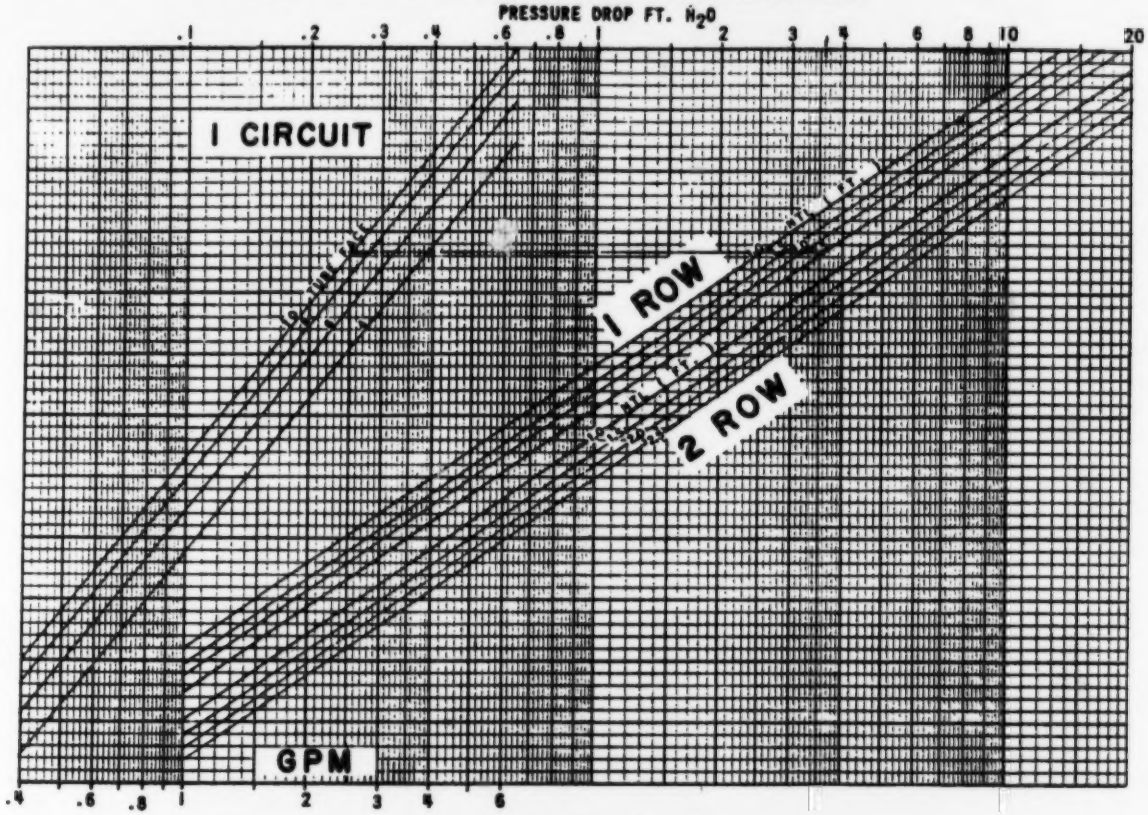


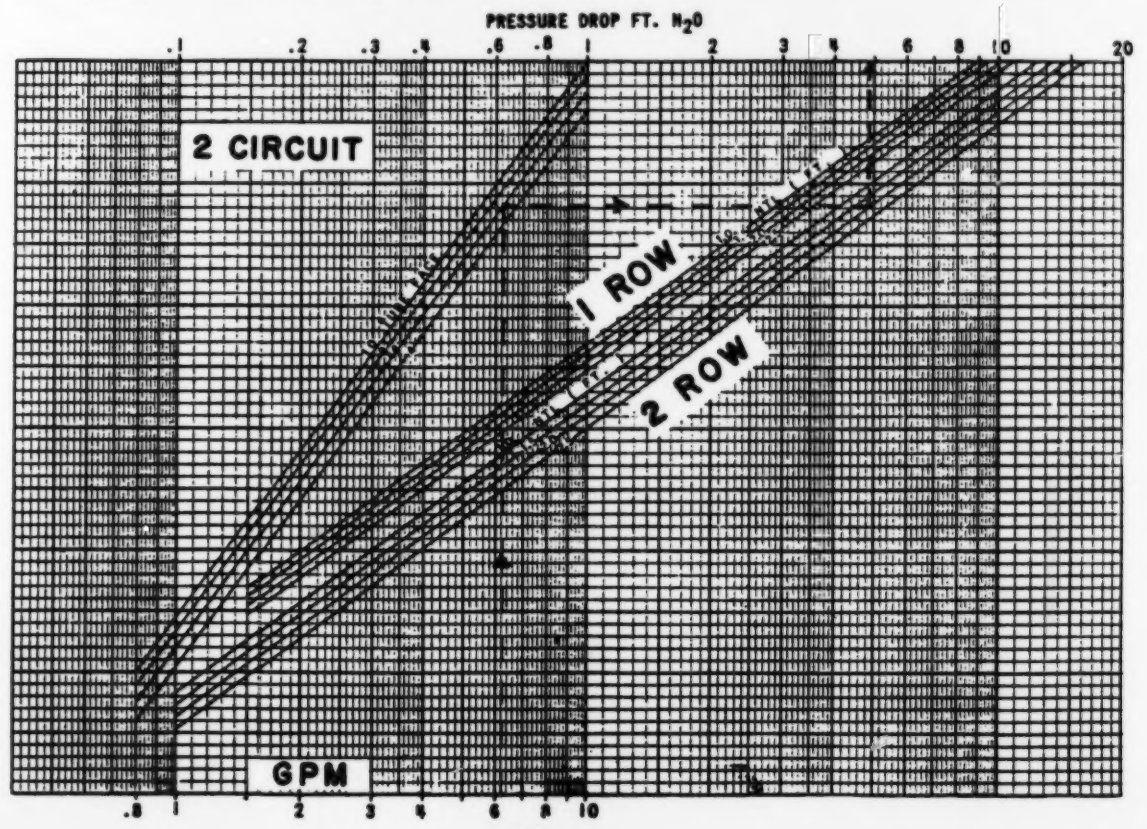
CURVE No. 4

140

WATER PRESSURE DROP TYPE MP FT. H_2O

150°F AVERAGE WATER TEMPERATURE





APPENDIX D

FLUE GAS CONDENSING WATER PREHEAT TANK

In order to arrive at a suitable water preheat tank design, it was necessary to consider a set of tubes passing through a tank of water as shown in Figure D.1. The design temperatures are also shown in Figure D.1.

It can be shown that the Reynolds number, Re_D, based on the hydraulic diameter, D, of the flue gas flow in the i^{th} tube is given by:

$$Re_D = \frac{4\dot{m}_i}{\mu p_i} \qquad (D.1)$$

where; $\dot{m}_i$ = flue gas mass flow rate in the i^{th} tube

μ = viscosity of the flue gas

p_i = perimeter of the flow area of the i^{th} tube

Knowing that if "n" tubes are used;

$\dot{m}_i = \dot{m}/n$ ($\dot{m}$ = 36 kg/hr = 79.4 lbm/hr, see Chapter 3)

and $p_i = \pi D$

equation D.1 can be rewritten as;

$$Re_D = \frac{4\dot{m}}{n\pi\mu D} \qquad (D.1)$$

The total rate of heat transfer from the flue gas to the water can be expressed as:

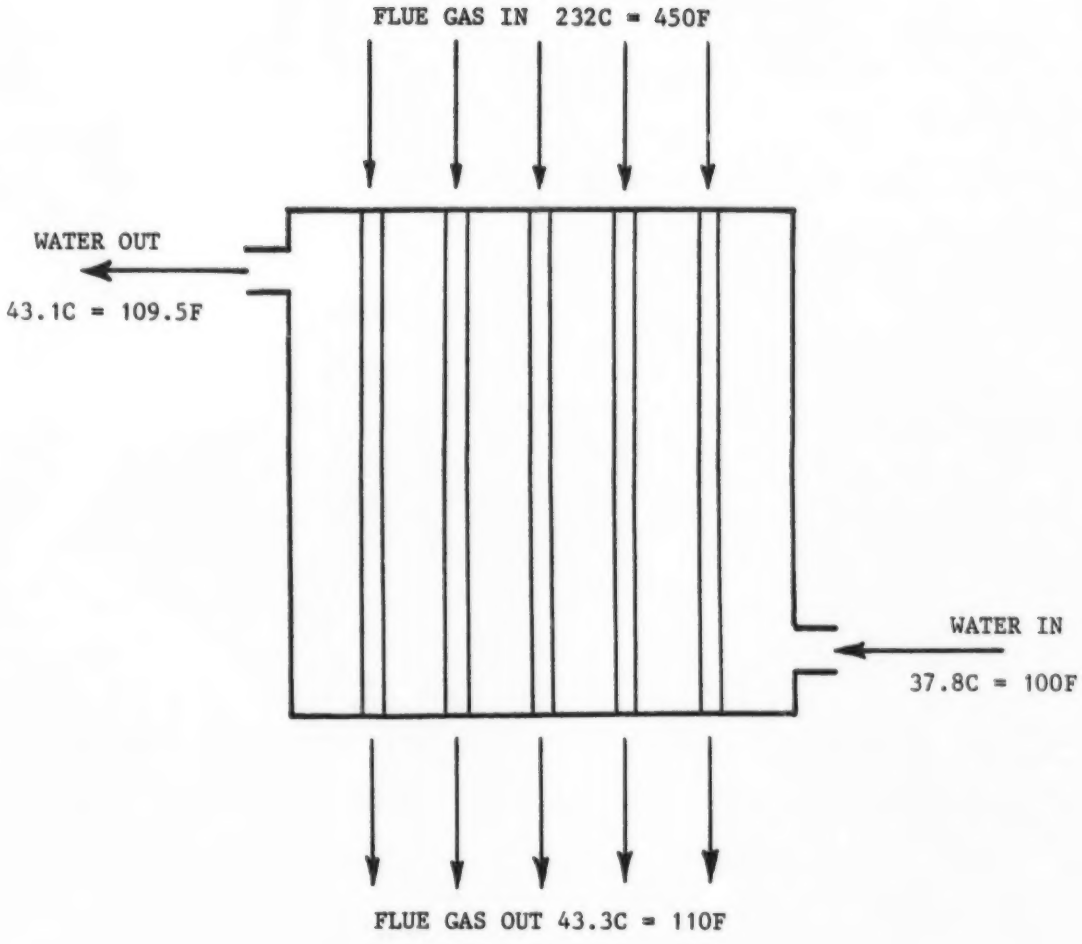


Figure D.1: Water Preheat Tank

$$\dot{Q} = AU(LMTD) = 15\% \text{ of burner input rate} \qquad (D.2)$$

where A = heat transfer area

U = heat transfer coefficient based on A

LMTD = log mean temperature difference between water and flue gas

Since the temperaure change of the water is relatively small, it is acceptable to use the LMTD as shown in equation D.2 even if the heat exchanger is not arranged in counterflow.

The heat transfer (tube wall) area can be expressed as:

$$A = n\pi DL \qquad (D.3)$$

where L = length of each tube

Neglecting the thermal resistance of the tube wall and of the water side film heat transfer coefficient, the U value quantifying the heat transfer is equal to the flue gas side film heat transfer coefficient, h_{FG}.

$$U \cong h_{FG} = Nu \frac{k}{D} \qquad (D.4)$$

where Nu = Nusselt number based on D

k = thermal conductivity of flue gas

When considering turbulent flow in a tube, the Nusselt number can be estimated using the correlation[28]:

$$Nu = 0.023\ Re_D^{0.8}\ Pr^{1/3} \qquad Re_D > 6{,}000 \qquad (D.5)$$

where Pr = Prandtl number of the flue gas

Specifying that Re_D = 6,000 in order that heat transfers from the flue gas to the tube wall by means of a turbulent flow mechanism and solving equation D.1 yields the result:

nD = 8.9 cm = 3.5 inches

where the flue gas viscosity is assumed to the same as that for air at the mean flue gas temperature ($\mu = 1.64 \times 10^{-3}$ lbm/ft sec).

Choosing a tube flow diameter of 1.27 cm (½ inch) indicates that seven tubes should be used.

Substituting Pr=0.7 and Re_D=6,000 into equation D.5 yields:

Nu = 21.5

Solving equation D.4 assuming that the thermal conductivity of the flue gas is the same as that for air yields:

U = 1.8 W/m² C = 10.3 BTU/hr ft² F

Using equation D.2:

$$A = 0.39\ m^2 = 13.9\ ft^2$$

where $$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} \cong 52.8C = 95F$$

and $\Delta T_1 = 450 - 109.5\ F$ $\Delta T_2 = 110 - 100\ F$

Therefore, from equation D.3, the length of each tube must be:

$$L = 4.6\ m = 15\ ft$$

This analysis does not account for the increased heat transfer rate in the last few feet of tubing which results from the condensation which takes place. Thus, a tube length of 4.6 m is probably a conservative estimate. If the heat exchange tubes are considered only to the point in their length where condensation starts (flue gas temperature ≅ 52.8C = 127F) then LMTD would be 68.4C (123.5F) and the tube length needed would only be about 3.7 m (12 ft).

The pressure drop experienced by the flue gas passing through the preheat tank was estimated using equation D.6 taken from reference 29.

$$\frac{\Delta P}{P} = \frac{G^2}{2g_c} \cdot \frac{v_1}{P} \{(K_c+1-\sigma^2) + 2(\frac{v_2}{\tau_1} - 1) + f \frac{A}{A_c} \cdot \frac{v_m}{v_1} - (1-\sigma-K_e)\frac{v_2}{v_1}\} \qquad (D.6)$$

G = mass flux = 58.9 lbm/hr in²

$$\frac{v_1}{P} = \frac{\text{specific volume of incoming flue gas}}{\text{pressure}} = 1.074 \times 10^{-2} \text{ ft}^5/\text{lbm lbf}$$

$$\sigma = \frac{\text{frontal heat exchanger area}}{\text{minimum flow area}} \quad \text{(assumed to be approximately zero)}$$

K_c = entrance flow coefficient = 0.5

K_e = exit flow coefficient = 1.0

$$\frac{A}{A_c} = \frac{\text{heat transfer area}}{\text{minimum free flow area}} = 1440$$

$$\frac{v_2}{v_1} = \frac{\text{specific volume of exiting flue gases}}{\text{specific volume of entering flue gases}} = 0.63$$

$$\frac{v_m}{v_1} = \frac{\text{mean specific volume of flue gases}}{\text{specific volume of entering flue gases}} = 0.73$$

$f = 0.007$ (from Moody diagram)

Solving equation D.6 yields:

$$\frac{\Delta P}{P} = 0.0075$$

which is equivalent to a flow head of about 7.6 cm (3 inches) H_2O.